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TRACK TRAIN DYNAMICS ANALYSIS AND TEST PROGRAM - LOCOMOTIVE DYNAMIC CHARACTERIZATION SUMMARY

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16. ABSTRACT

This report provides an analysis of locomotive mechanical characteristics, track perturbations, and operational characteristics involving experimentally determined suspension system parameters. Report makes comparative evaluations among locomotive designs, especially suspension bearings, shock absorbers, pads, and two- and three-axle trucks.

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FOREWORD

This report, prepared by Martin Marietta Denver Aerospace, presents a summary of all locomotive truck testing conducted under Contract NAS8-29882. The testing was conducted at Martin Marietta's structures test laboratory from mid-1976 through mid-1980. The contract is administered by the National Aeronautics and Space Administration, George C. Marshall Space Flight Center, Huntsville, Alabama, under the direction of Mr. Ismail Akbay.

ACKNOWLEDGEMENTS

The Program Manager would like to acknowledge several individuals and organizations who have contributed to the successful completion of the testing phase of this contract.

Mr. Ed Lind	:	Southern Pacific Railroad, arranged for many of the truck test articles.
Electro-Motive Division of General Motors	:	Supplied the "New" HT-C and E8 truck test articles. Also contributed information for use in this report.
Amtrak	:	Supplied the E60, and modified GPSS truck test articles.
Southern Pacific Transportation Company	:	Supplied the "standard" HT-C truck test article.
Southern Railroad	:	Supplied the GPSS truck test article.
Chesapeake & Ohio Railroad	:	Supplied the Flexicoil truck test article.
Burlington Northern Railroad	:	Supplied the U30 truck test article.
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Mr. Ismail Akbay	:	NASA, Marshall Space Flight Center, provided contract administration.
Dr. George Morosow	:	Martin Marietta, was Program Manager during the majority of the contract.
Mr. Pete Abbott	:	Martin Marietta, acting as Technical Director and consultant has contributed invaluable technical advice and direction during the testing and data analysis.
Mr. Dick Vigil	:	Martin Marietta, was test engineer responsible for acquiring and formatting the test data.
Mr. Al Nemes Mr. John Bradford Mr. Nord Hjerleid	:	Martin Marietta, served as test conductors responsible for test setup, instrumentation, and everyday operations.

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1.0 INTRODUCTION

1.1 Background

The demands on the country's rail transportation systems have steadily risen over the past few decades. In accommodating themselves to shippers' requirements, the railroads have introduced a number of new locomotives, creating concern about higher derailment risk. One of the prime targets for extensive investigation has been the locomotive truck.

Following several Amtrak derailments, severe speed restrictions were placed on locomotives having a particular type of truck. These restrictions had such a drastic effect on Amtrak schedules that the FRA initiated, in a very time responsive manner, several large studies to experimentally analyze and compare locomotive truck characteristics. The FRA's objective was to assure all concerned that an acceptable level of safety existed, while gathering data upon which to make rational decisions regarding operational restrictions.

Prior to the locomotive derailment problems, Martin Marietta had been awarded a contract to experimentally characterize the standard three-piece freight truck. Because of Martin Marietta's success in determining the dynamic characteristics of freight car trucks, they were awarded a contract to apply the same type of methodology in characterizing locomotive trucks. This report summarizes the test program which followed, including the results of tests on eight locomotive trucks.

1.2 Project Accomplishments

The locomotive truck performs a number of essential functions. The truck: 1) attenuates the magnitude of impulsive and periodic forces arising from track imperfections (which are transmitted to the carbody and lading), 2) damps motions of the carbody excited by rail irregularities, 3) maintains adequate vertical wheel loads to guard against derailment in the presence of lateral forces, 4) transmits traction and braking forces, and 5) guides the carbody around curves.

Recent advances in locomotive truck design have made the task of evaluating truck performance even more complex. Design changes range from add-on devices to new truck configurations that are based on the better understanding of vehicle dynamics gained from analysis, laboratory experiments and road tests.

Improvements to any piece of hardware requires detailed knowledge of what exists. Hence, characterization must precede and form the basis for any engineering advancement. Since railroads tend to be pragmatic, their preferred procedure for evaluating truck operating performance has been by road test. This study has developed a laboratory test procedure that approximates the real world and includes many of its complexities. The new laboratory procedure will greatly reduce the difficulties of interpreting data due to unforeseen interactions within the truck.

Test conditions were selected to measure, either directly or indirectly, the load deflection characteristics which correspond to the relative motions or degrees of freedom of a truck.

The specific degrees of freedom evaluated were:

- Bolster-frame relative roll (secondary roll);
- Frame-axle relative roll (primary roll);
- Bolster-frame relative vertical displacement (secondary vertical);
- Frame-axle relative vertical displacement (primary vertical);
- Bolster-frame relative lateral displacement (secondary lateral);
- Frame-axle relative lateral displacement (primary lateral); and
- Frame-axle relative yaw (primary yaw).

An important consideration in the measurement of these characteristics is the effect of natural steady loads acting on the truck. These loads include the locomotive weight (one-half for each truck) and longitudinal forces due to braking or traction. Simulation of these loads was included in the laboratory test procedure.

It is difficult to simulate operating conditions in a laboratory without any test induced effects. However, through an intensive evaluation of the raw data, these undesirable effects were successfully eliminated.

The final phase of the study included development of an analytical computer model for evaluating the operational safety of six-axle locomotives. The analytical model was structured to accept hardware-oriented suspension test data from the test program, as well as truck and locomotive geometry and mass properties, and definition of track geometry defects. The model was used to perform parametric studies on the HT-C locomotive truck, in order to highlight truck parameters important to operational safety. In addition, a limited comparative analysis was conducted on HT-C, E8 and U30 locomotives to demonstrate the analytical tool's flexibility. The analytical efforts are detailed in Reference 1.

1.3 Future Prognosis

There are few standards by which an individual locomotive truck design may be judged. Certainly, the AAR Mechanical Division does not address dynamic response characteristics of locomotive trucks, or any of its components, in any detail. It is apparent that the next step in this study should be the establishment of a laboratory test and analysis procedure for determining the acceptability of new or modified truck designs, taking into consideration wear and maintenance programs.

More explicitly, the acceptance program should incorporate the following steps:

a) Experimental determination of truck suspension characteristics;

b) Analysis of the design, based on test data, to identify any potential operational safety problems;

c) Accelerated wear and fatigue tests to determine projected maintenance costs and failure frequency;

d) Limited road testing (e.g., Pueblo FAST testing);

e) Review by industry and government using the following criteria:

- Safety
- Reliability (failure rate)
- Economics (life cycle analysis); and

f) Production for sale to industry.

Establishment of a common facility, available for industry use, to conduct these acceptance tests has several benefits:

- Established procedures/techniques;
- Cost and schedule considerations;
- Qualified personnel; and
- Accessibility of facility and test equipment.

Test uniformity and reproducibility make a single facility highly desirable, allowing for valid comparisons between trucks, hence, making test results more readily accepted by the industry.

Railroads would certainly benefit from such an acceptance procedure. However, the increased cost of such a procedure must be weighed against the projected benefits before industry support can be obtained.

2.0 TEST SUMMARY

This chapter summarizes the test programs conducted under this contract from 1976 to 1980. During this period, eight locomotive trucks were tested (six three-axle trucks and two two-axle trucks), along with element tests on some key components of the truck suspension system. A test methodology has been developed in which the truck is tested as a system, using quasi-static techniques to establish key suspension parameters. The test data obtained clearly define the hardware physical behavior and, hence, can be used effectively in improving the quality of analytical simulations.

2.1 Truck Test Program

The locomotive truck provides the locomotive carbody suspension system, in addition to transmitting traction and braking forces between the carbody and track. Figure 2-1 is a photograph of a General Electric P-30CH locomotive, showing a typical carbody/truck configuration. There are two trucks per carbody. Depending on application, trucks have either two or three axles with two to three electric or diesel electric drive motors (some three-axle trucks have only two drive motors).

The truck consists of several basic "elements" connected by mechanisms such as springs, dampers, and bearings. These elements are:

1. Bolster
2. Spring plank assembly*
3. Frame
4. Wheel/axle set
5. Electric or diesel electric motors

There are other pieces of hardware on the trucks (e.g., brake rigging, plumbing), but these are not explicitly pertinent to the dynamic behavior of a truck.

There are two basic suspension designs: 1) the "standard" design#, and 2) the "swing hanger" design. The difference in the two designs is primarily in the connection of the bolster to the frame. In the standard design, the bolster and frame are connected via compression springs (rubber or steel). The swing hanger design employs a lateral pendulum suspension arrangement, between the bolster and frame. Figure 2-2 is a photograph of a standard design truck: Three-axle GE U30C**. Figure 2-3 is a photograph of a swing hanger design truck: Two-axle EMD GPSS†.

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*The spring plank assembly is only present in trucks having a swing hanger (pendulum) lateral suspension system.

**GE = General Electric.

†EMD = Electromotive Division of General Motors.

#Also called floating bolster design.

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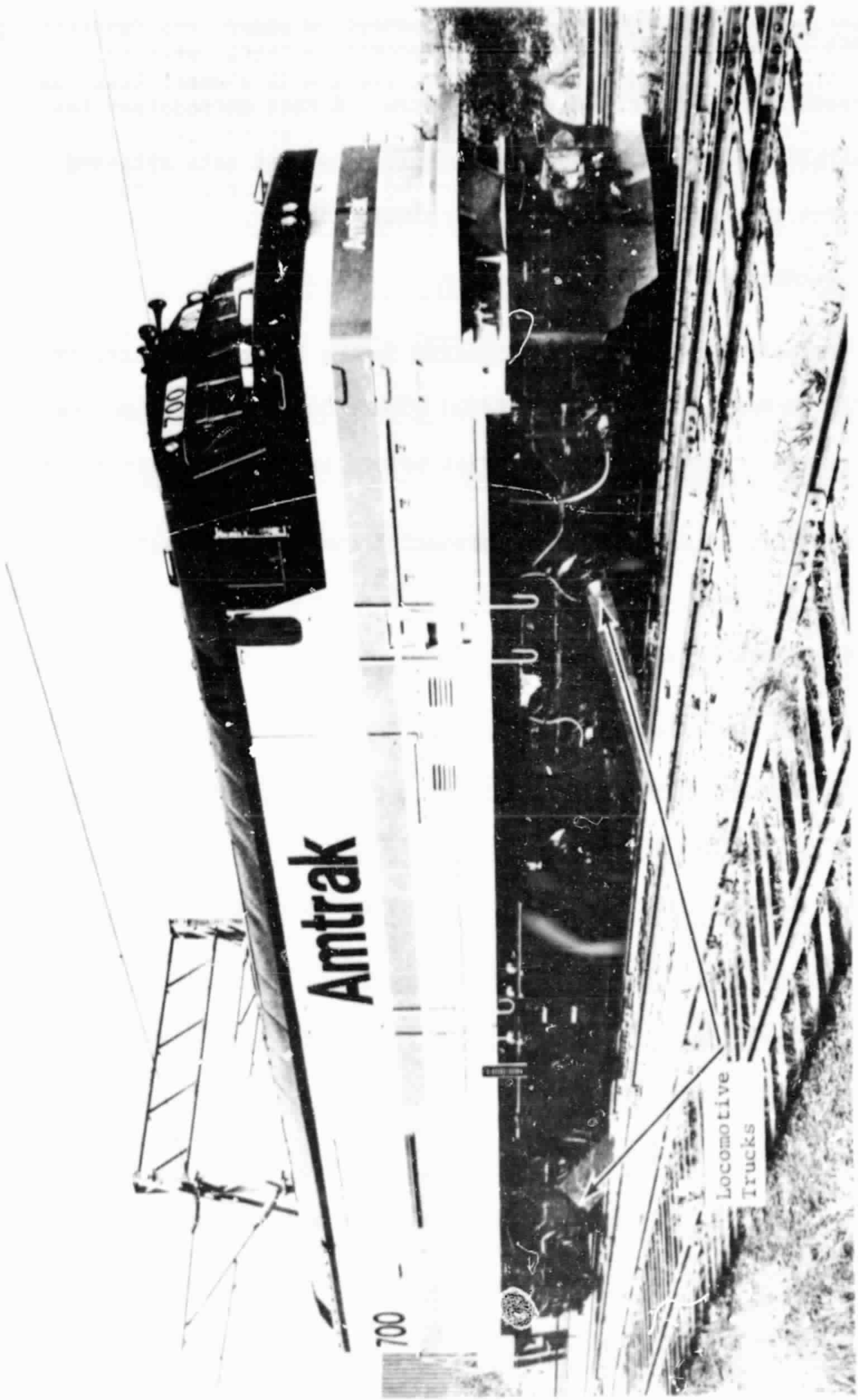

Figure 2-1 General Electric P-30CH Locomotive

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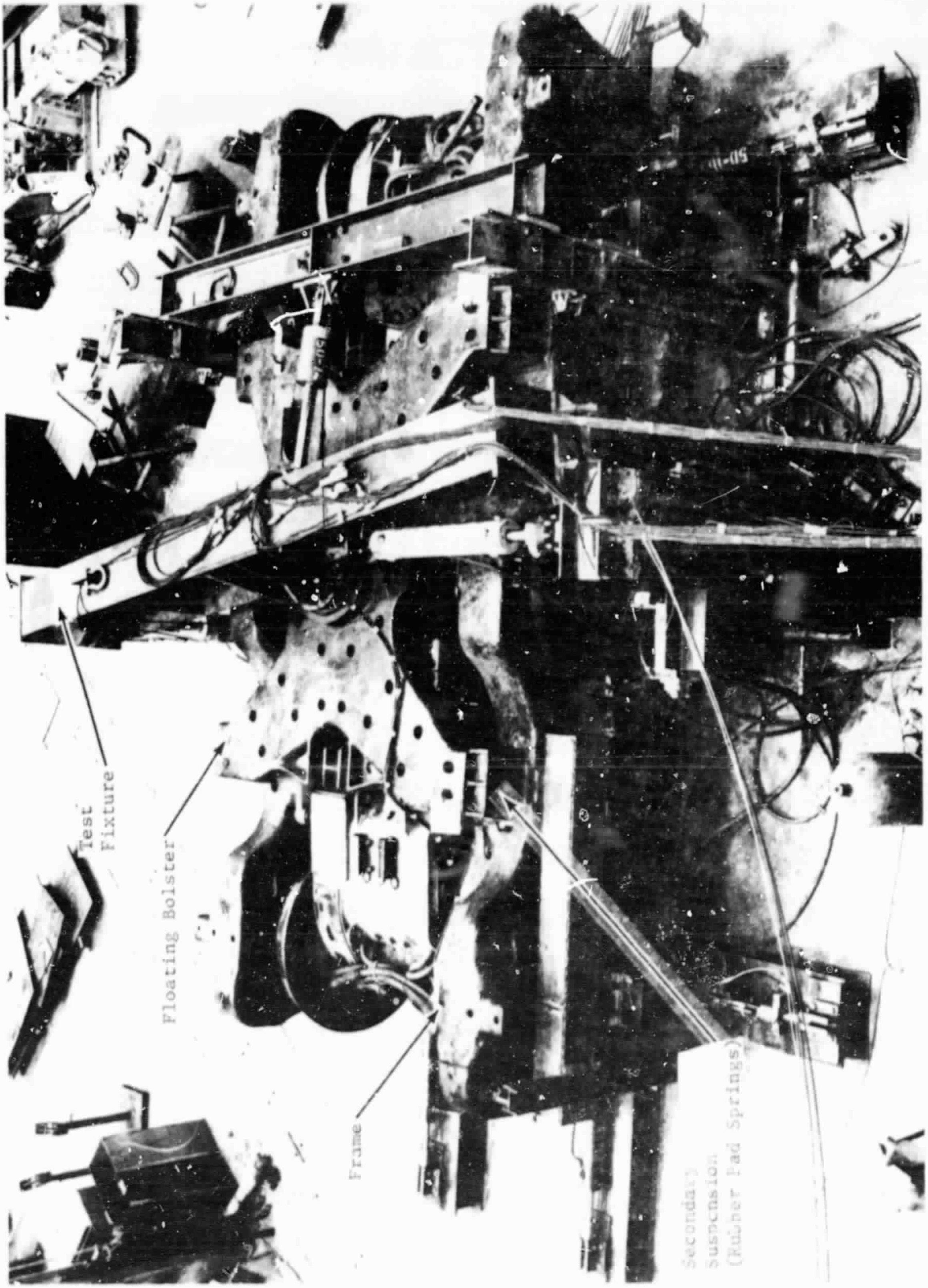

Figure 2-2 Typical Standard Truck: 3-Axle General Electric U30C

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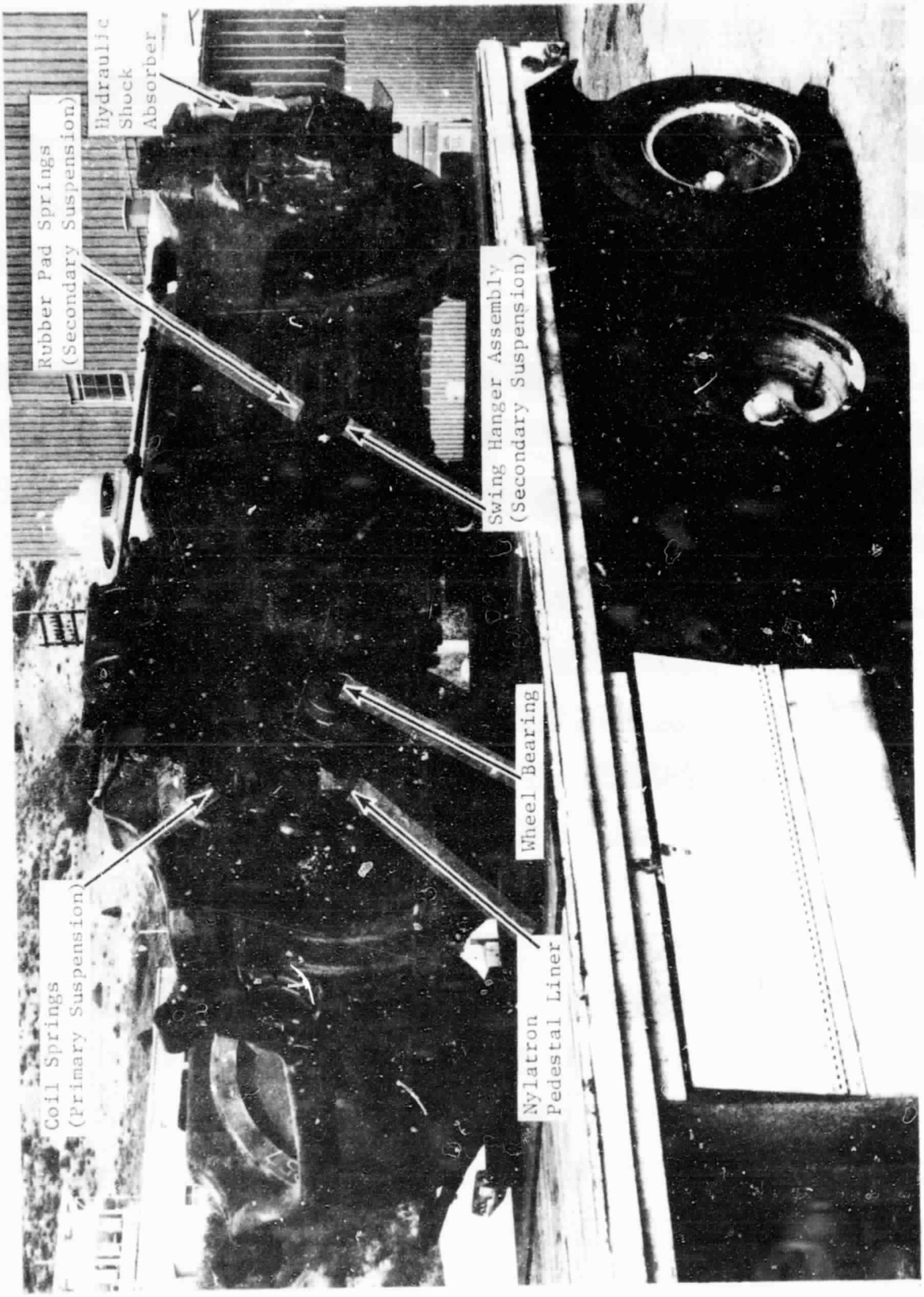

Figure 2-3 Typical Swing Hanger Truck: 2-Axle General Motors GPSS

The truck suspension system can be divided into two parts: secondary and primary. The secondary suspension system is between the bolster and the frame. The primary suspension system is between the frame and the axles. Table 2-1 delineates the components of the secondary and primary suspension systems for both the standard and swing hanger designs. Figure 2-4 is an artistic idealization of a truck's primary suspension system. Figure 2-5 is an idealization of the secondary suspension system for both standard and swing hanger design trucks.

Most analytical modeling techniques assume that the truck "elements" are rigid and, hence, any elastic deformations of these "elements" are not significant to the dynamic behavior of the truck. The significant relative displacements in the truck occur in the connections between "elements". It was the objective of the test program to measure the physical characteristics of these connections.

2.1.1 Test Approach

The various interaction forces within a truck may be classified as being sensitive to acceleration, velocity, or displacement. Acceleration sensitive forces are influenced by the truck's mass and inertia properties. The velocity sensitive forces are damping forces, due to rubber pad springs and auxiliary devices such as shock absorbers. The displacement sensitive forces are related to the stiffness properties of connections.

The test approach was designed to measure the displacement sensitive characteristics of the truck including friction. Although friction between elements is sensitive to the direction of their relative velocity, it is very nearly independent of the magnitude of the velocity. Mass properties and damping descriptors, in general, were not measured; they were calculated or obtained experimentally from element tests.

Test conditions were chosen to measure, either directly or indirectly, the load/deflection characteristics representative of the degrees of freedom which might be simulated in a truck analytical model. Table 2-2 tabulates the four test conditions employed and the information acquired from each test.

Trucks were mounted on a test fixture, during testing. The fixture (Figure 2-6) supported the truck's wheels on slide plates which rested on lubricated, machined fixture surfaces. The slide plates were restrained with hydraulic actuators to react the wheel loads. The fixture included provisions for applying vertical, lateral, and longitudinal (tractive effort/braking) loads to the truck bolster center plate. The restraining actuators, on the wheel slide plates, were also used to apply axle yaw forces. Figure 2-7 shows the overall test setup with an EMD GPSS (two-axle) truck installed. The fixture was housed in a cold enclosure, during several of the tests, to allow low temperature testing (0^{o}F) on trucks with rubber suspension elements. Figure 2-8 is a close-up photograph of the loading interface between the load beam and bolster center plate. The knife-edge adapter shown was used to load swing hanger trucks, in order to prevent unrealistic test related constraints.

Table 2-1 Truck Suspension Details

SUSPENSION SYSTEM	DESIGN	CONNECTION	DIRECTION	EXAMPLE MECHANISMS
Secondary	Swing Hanger	Bolster to Spring Plank	Vertical	Leaf Springs, Rubber Pad Springs
			Lateral	Rubber Pad Springs, Rigid Mechanical Stops
		Spring Plank to Frame	Lateral	Swing Hanger Assembly (Pendulum Action)
		Bolster to Frame	Longitudinal	Tractive Effort Stops (Rigid), Friction Mechanisms
	Standard	Bolster to Frame	Vertical & Lateral	Leaf Springs, Coil Springs, Rubber Pad Springs
			Longitudinal	Tractive Effort Stops (Rigid), Friction Mechanisms
Primary	Swing Hanger & Standard	Frame to Axle	Vertical	Coil Springs, Shock Absorbers
			Lateral	Bearings
			Longitudinal	Coil Springs With Stop When Journal Box Contacts Pedestal Liner

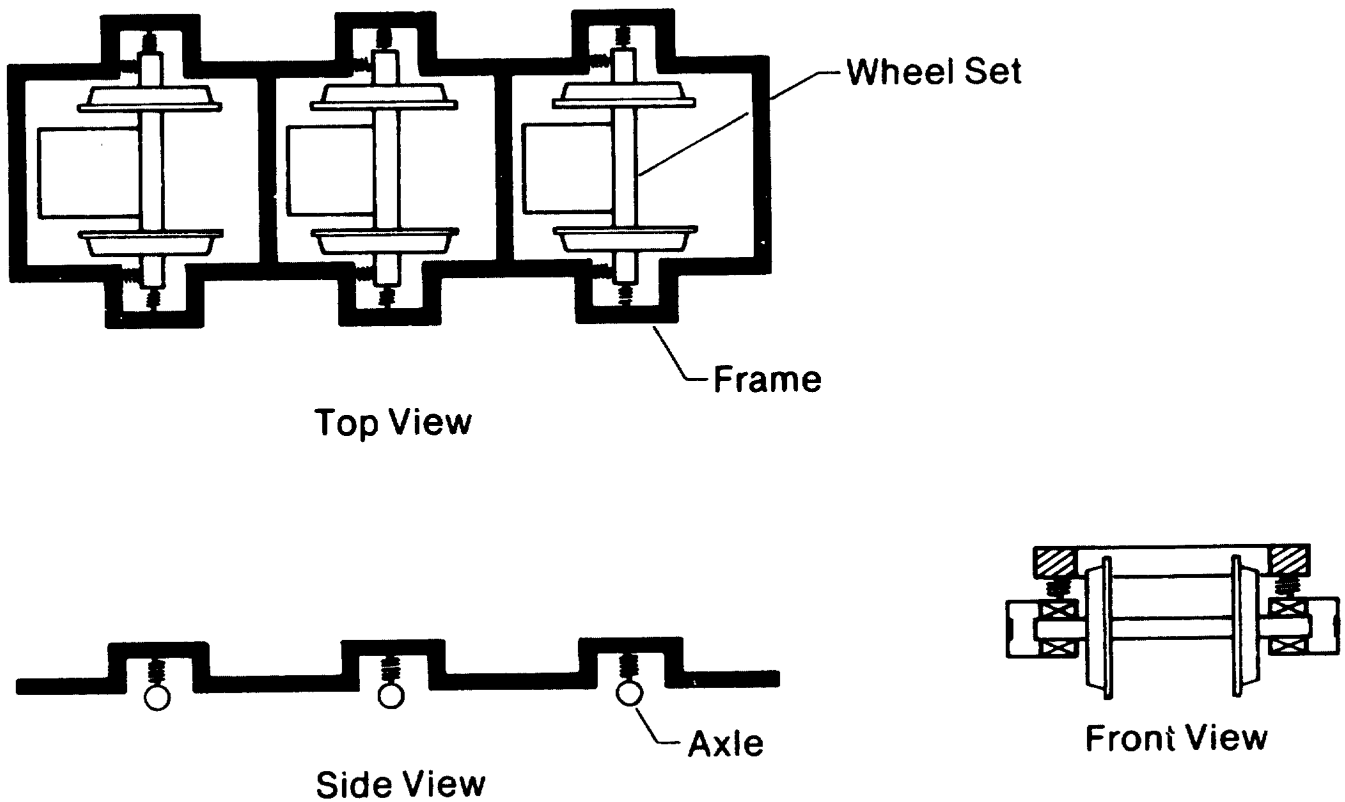

Figure 2-4 Suspension Idealization: Primary Suspension - Frame/Wheelset Connection

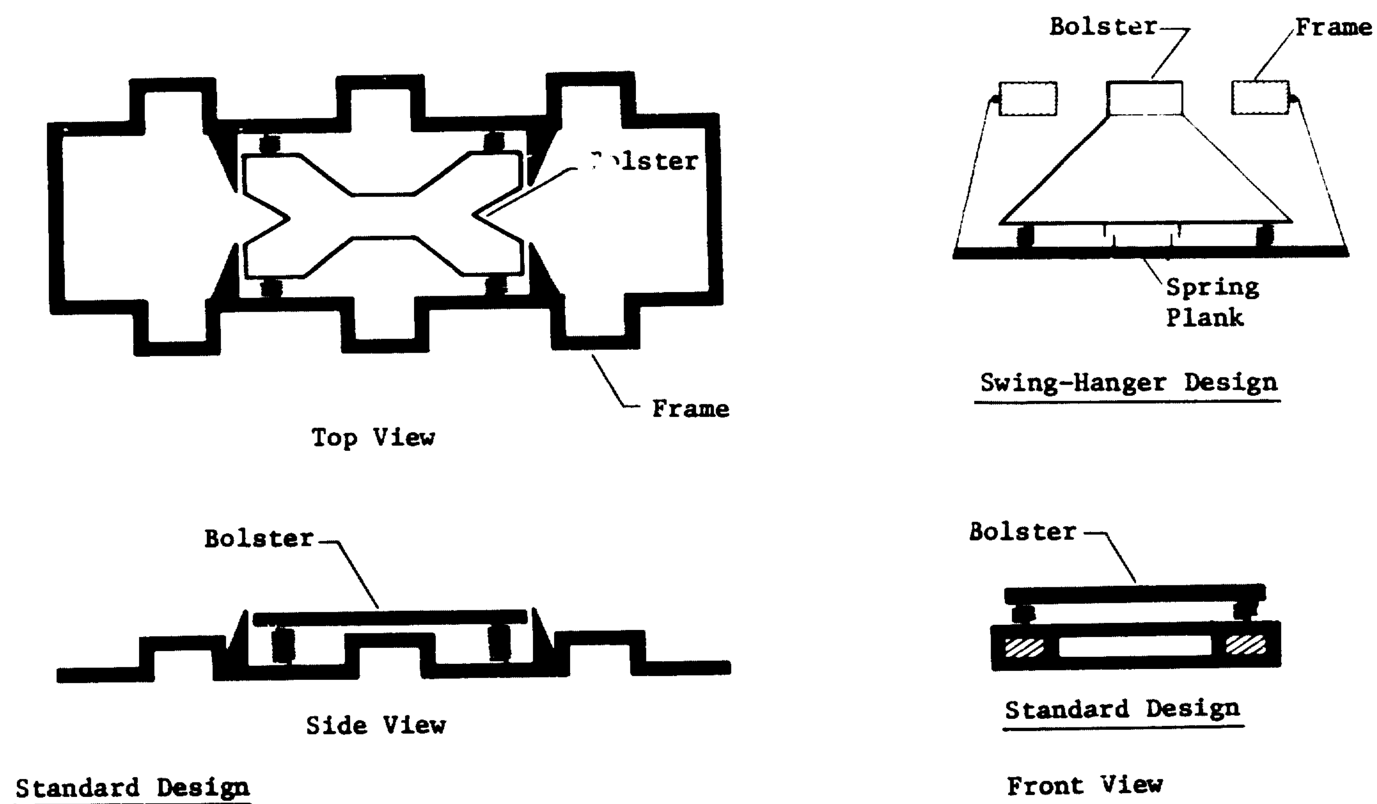

Figure 2-5 Suspension Idealization: Secondary Suspension - Bolster/Frame Connection

Table 2-2 Matrix Relating Information Acquired to Test Condition

TRUCK DEGREE OF FREEDOM	TEST CONDITION			
	1 VERTICAL	2 LATERAL	3 AXLE YAW	4 BOLSTER YAW (WET/DRY)
Bolster Roll	X	-	-	-
Frame Roll	X	-	-	-
Bolster Lateral	-	X	-	-
Frame Lateral	-	X	-	-
Carbody Bolster Yaw	-	-	-	X
Frame Yaw	-	X	-	-
Axle Lateral	-	X	-	-
Axle Yaw	-	-	X	-

X = Information obtained from test condition.

- = No information obtained from test condition.

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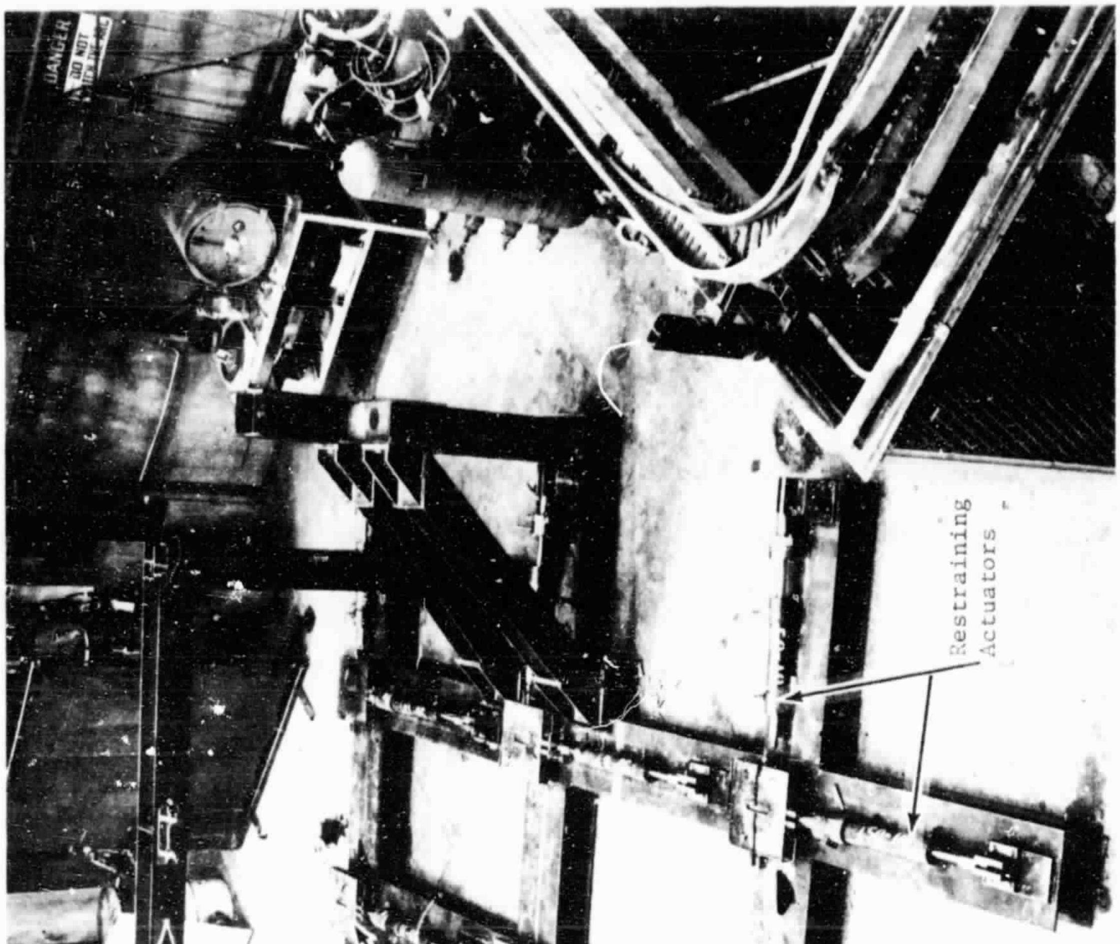

Figure 2-6 Locomotive Truck Test Fixture

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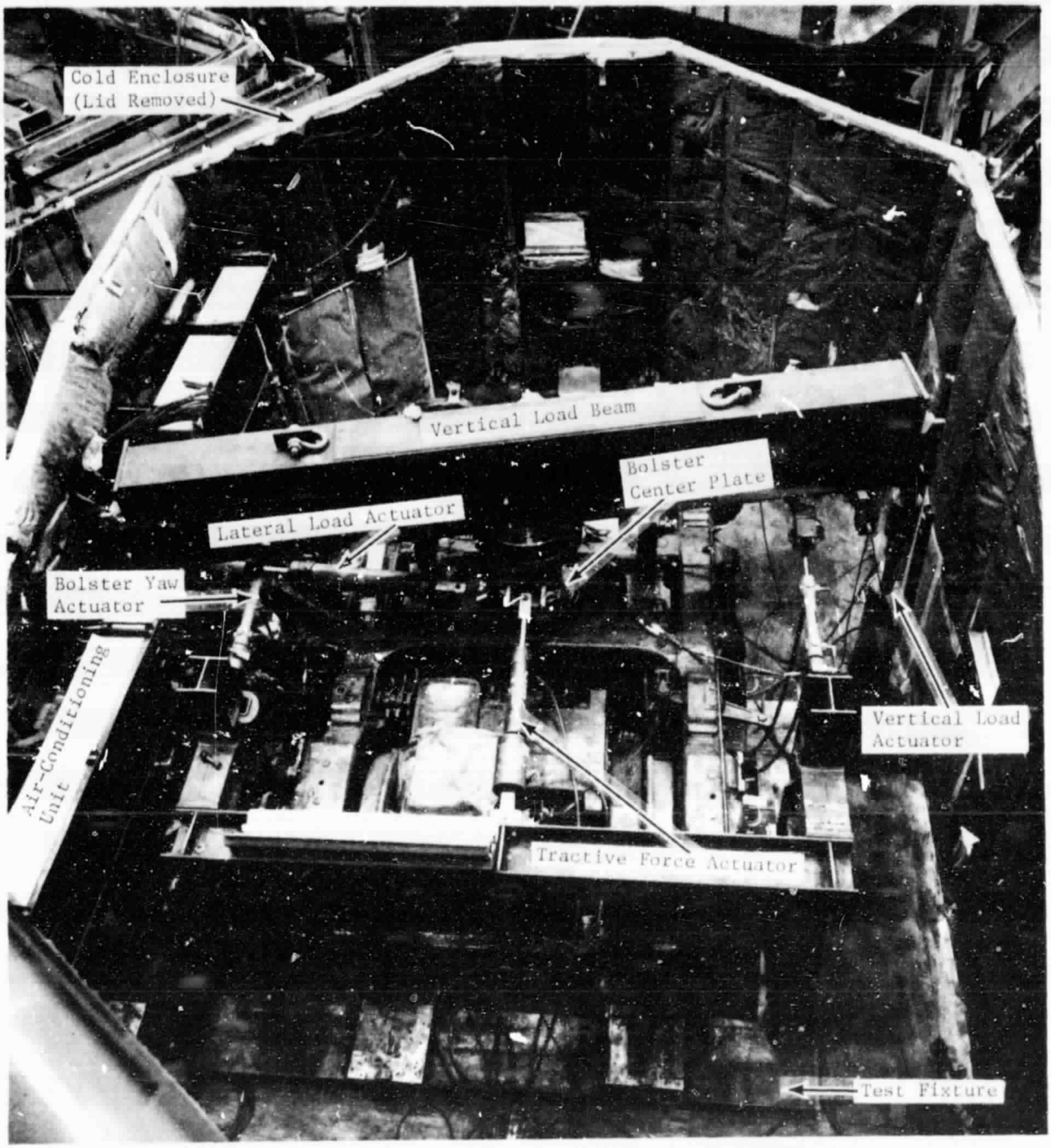

Figure 2-7 Martin Marietta Locomotive Truck Test Facility with 2-Axle GPSS Truck Installed

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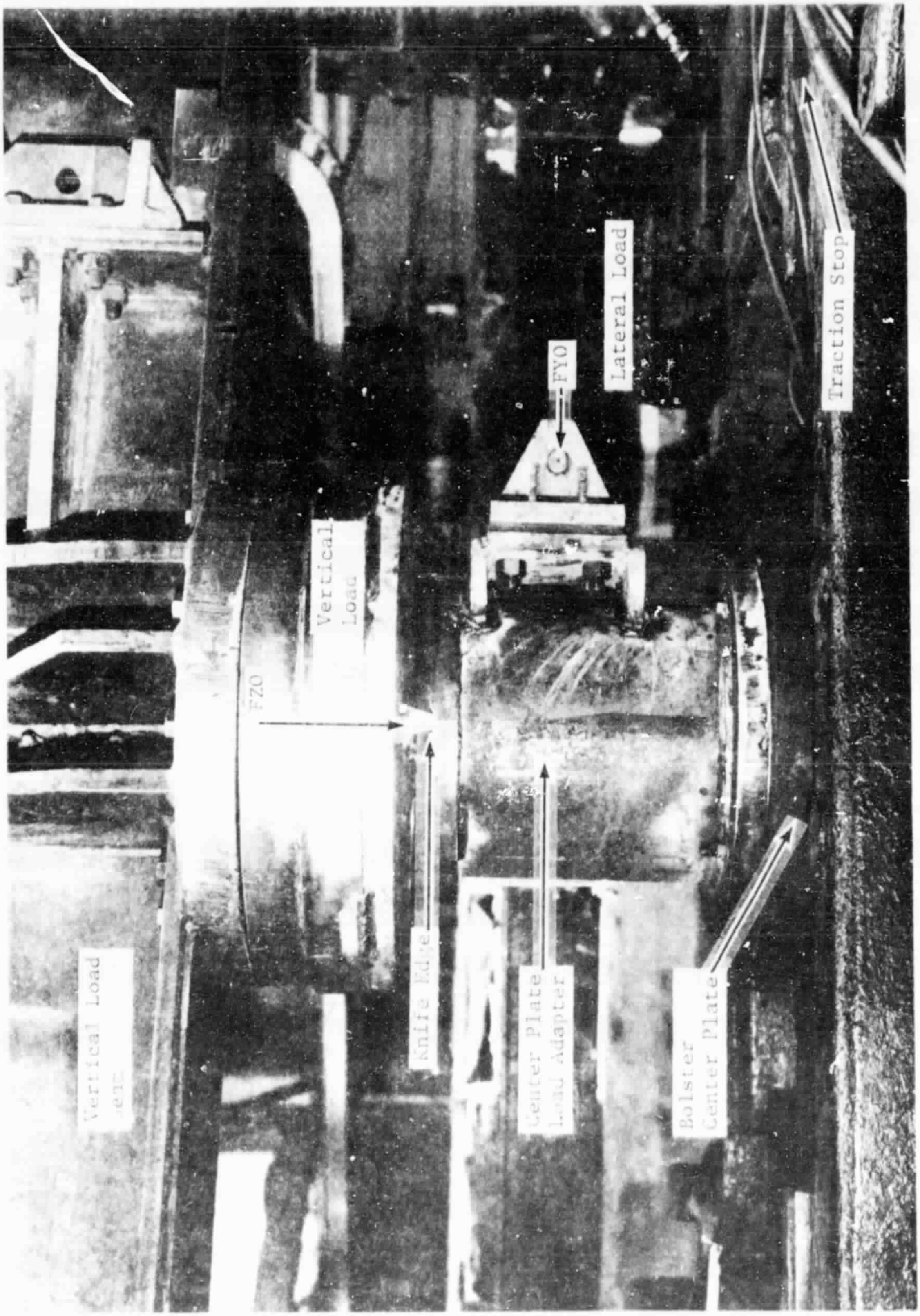

Figure 2-8 Bolster Center Plate Loading Detail

An important consideration in the measurement of a truck's suspension characteristics is the effect of natural steady loads acting on the truck. These include locomotive weight (one half for each truck) and longitudinal forces, due to braking and traction. The test conditions accounted for these loads. Relative roll and yaw characteristics were determined indirectly by measuring the vertical and lateral displacement characteristics and then calculating the roll and yaw characteristics.

Table 2-3 presents the general test matrix used for all trucks. Specific loading values were tailored to the truck being tested. Figure 2-9 pictorially shows the loading conditions for the four test conditions employed.

Table 2-3 Suspension Characterization Test Matrix

	APPLIED AT BOLSTER CENTER PLATE				
TEST CONDITION	VERTICAL LOAD	TRACTIVE FORCE LOAD	LATERAL LOAD	BOLSTER YAW LOAD	AXLE YAW LOAD
1) Vertical#	Cyclic*	Constant	-	-	-
2) Lateral	Constant	Constant	Cyclic*	-	-
3) Axle Yaw	Constant	-	-	-	Cyclic*
4) Bolster Yaw	Constant	Constant	-	Cyclic*	-

* Cyclic loads were applied at ≈ 0.25 Hz.

\# Test 1 was run with and without the truck's brakes set to determine friction effects.

2.1.2 Data Acquisition and Interpretation

Test loads were applied via hydraulic load actuators, and strain gage load cells, in series with the actuators, were used to measure test loads. Both potentiometer and linear variable differential transformer (LVDT) deflection transducers were used to measure relative deflection, within the truck's suspension system. The instrumentation system accuracy was measured to be approximately 2% of full-scale calibration. Figure 2-10 shows a typical instrumentation installation.

The testing and data acquisition/reduction was controlled via an HP5451C, microcomputer based, test/analysis system. Figure 2-11 delineates a typical test scenario. The HP5451C system is able to simultaneously output excitation signals to the test specimen and acquire transducer response signals. The transducer outputs (from load and deflection transducers) were digitized in

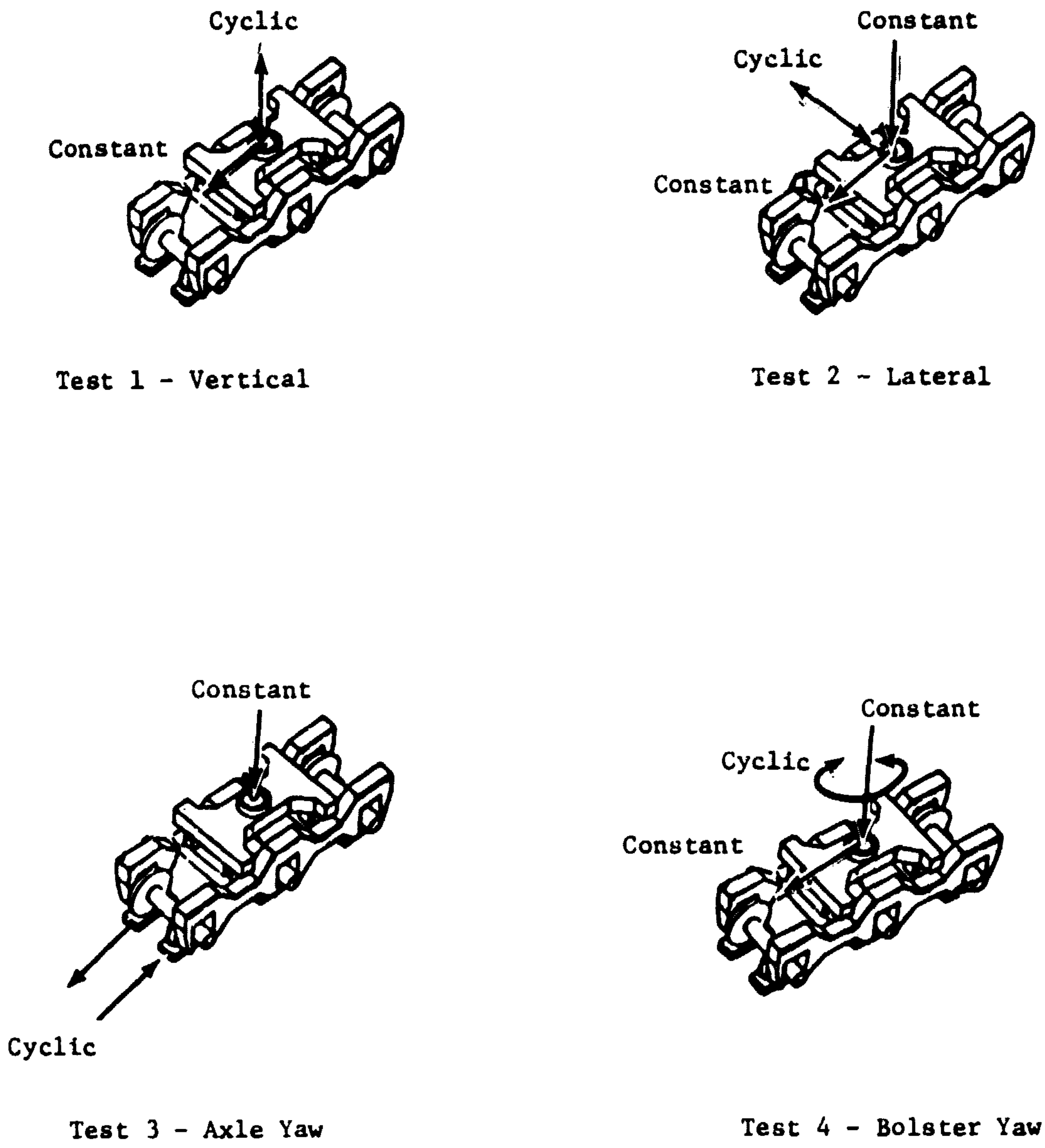

Figure 2-9 Truck Test Loading Conditions

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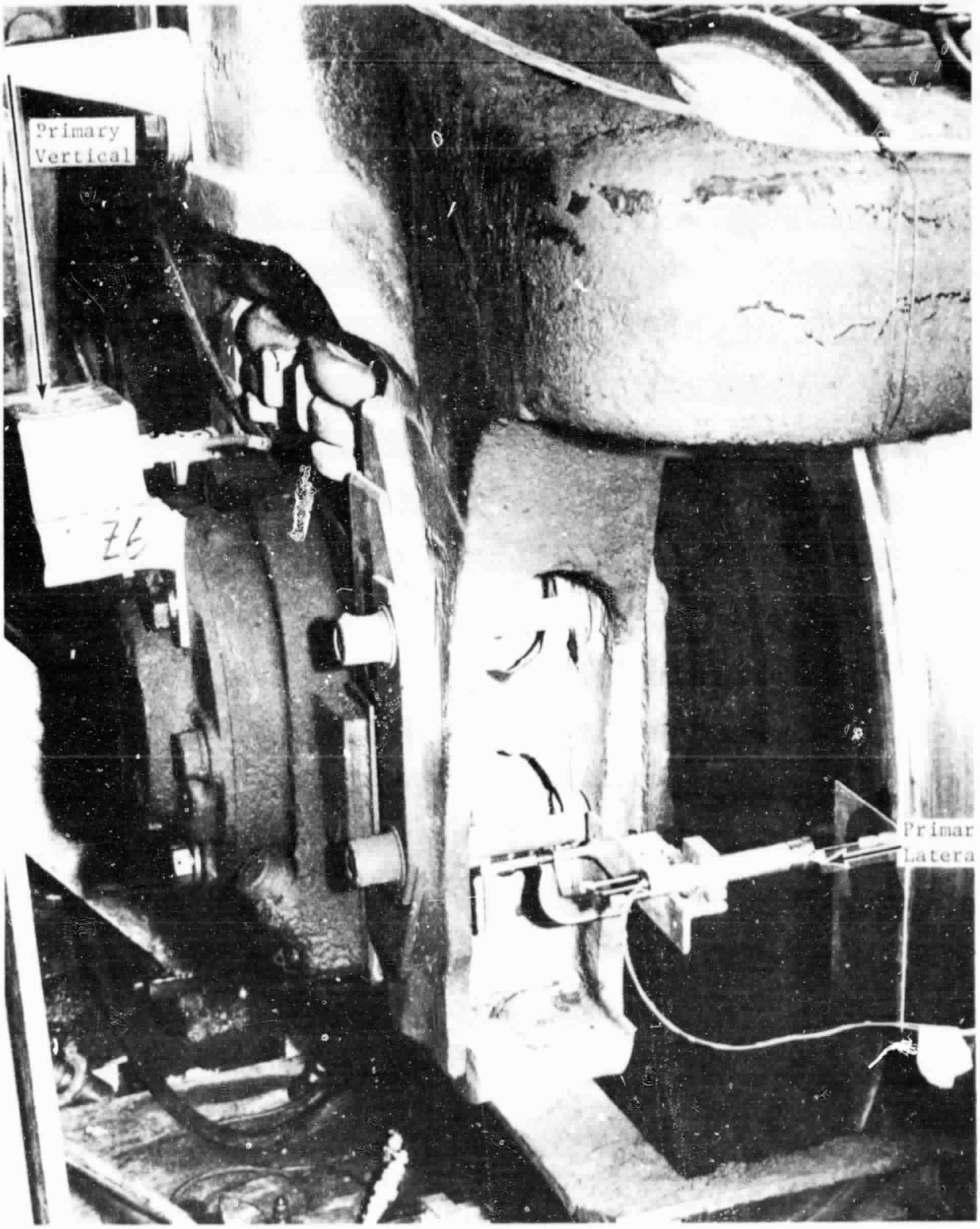


Figure 2-10 Typical Instrumentation Installation

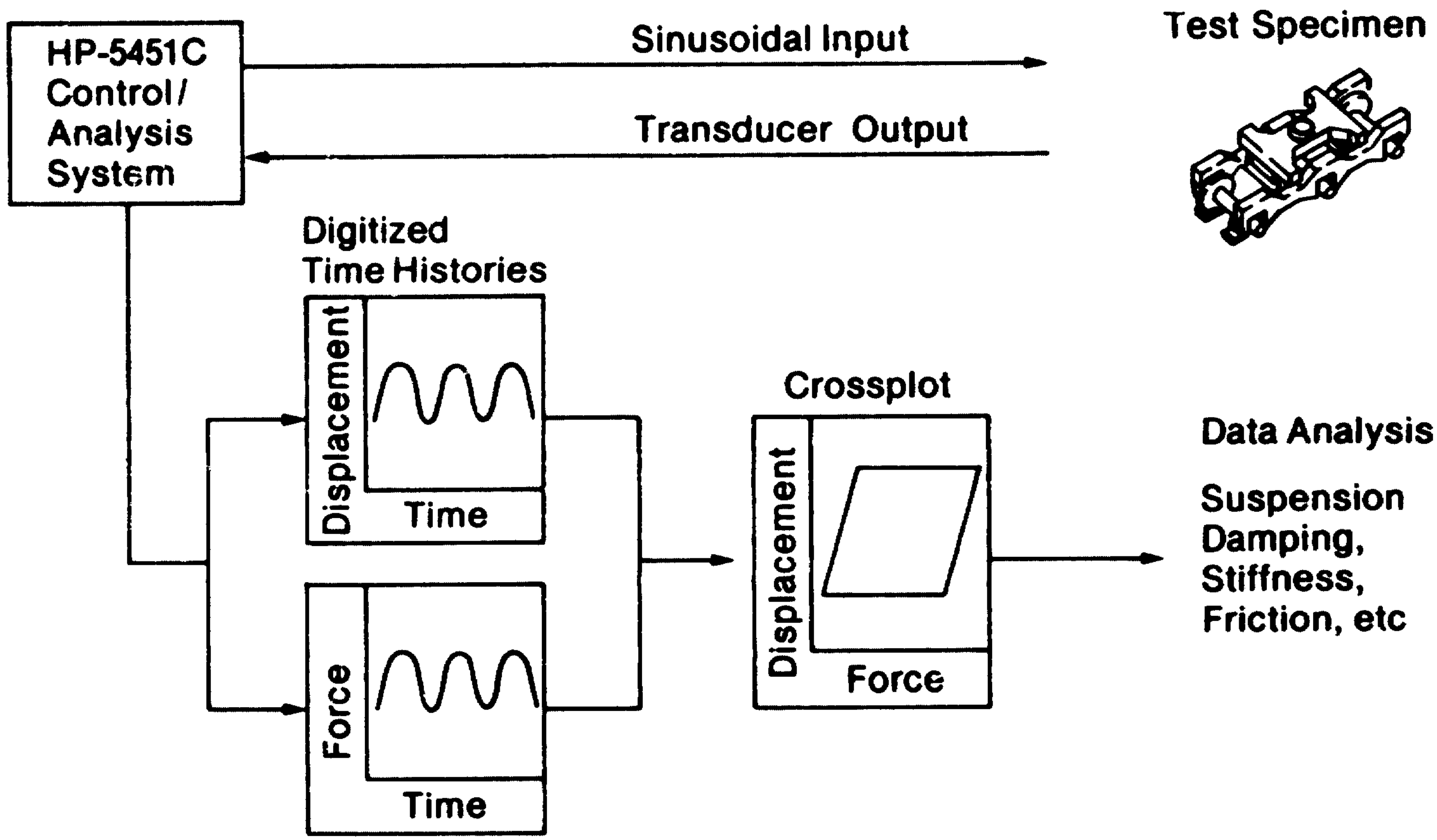


Figure 2-11 Typical Truck Test Scenario

real time and stored on magnetic storage disks. The computer then operated on the digital data converting them to engineering units and generated cross plots of force and displacement. The resulting hysteresis plots were evaluated to determine suspension stiffness and friction characteristics. Interpretation of typical load/deflection hysteresis characteristics found in a truck suspension system are discussed below.

The most common load/deflection characteristic is that of a linear spring (Figure 2-12a). This relationship is uniquely defined by the single parameter, K, which is the slope ($\frac{\Delta \text{ load}}{\Delta \text{ deflection}}$) of the load versus deflection curve. This type of relationship is characteristic of such suspension components as coil and leaf springs.

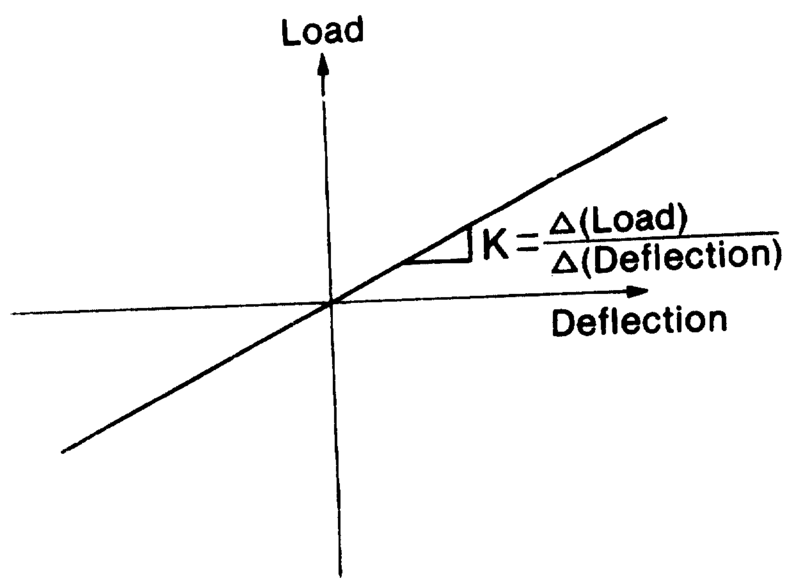
A second common characteristic is bilinear stiffness (Figure 2-12b). The connections in a truck, where this characteristic would be expected, are those which allow some motion, prior to hitting a deflection limiter ("stop"). This characteristic requires three parameters: K_1, K_2, and δ. The parameter δ defines the break point where the slope of the curve changes from K_1 to K_2.

The characteristics shown in Figure 2-12 are for suspension elements with no damping or friction. Most locomotive truck designs rely heavily on friction to provide suspension damping. Although the sense of the friction force is determined by the direction of the velocity vector (the friction force retards relative motion), the magnitude of the friction force is nearly independent of the magnitude of the velocity vector. Figure 2-13 shows the hysteresis curve for pure friction. The area within the hysteresis loop is a measure of the energy dissipated per cycle of motion.

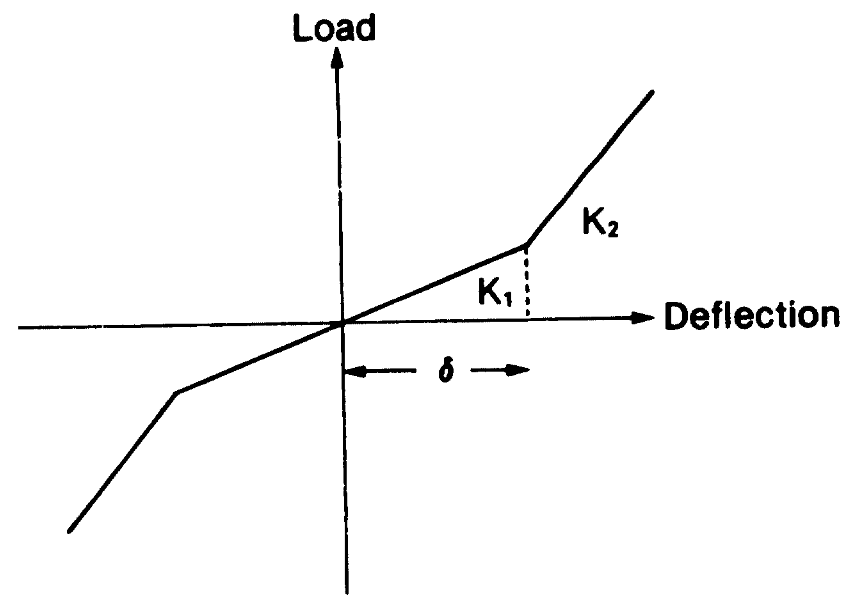
Combining linear and bilinear stiffness with friction results in the hysteresis characteristics shown in Figure 2-14. In addition to friction, a truck has other damping sources. Viscous damping is characteristic of rubber pad suspension springs and external hydraulic shock absorbers. Figure 2-15 shows the hysteresis characteristic of a linear spring combined with viscous damping. The magnitude of the viscous damping force is a function of the loading rate (velocity). The testing employed a fairly low loading rate and, consequently, viscous damping effects were not perceptible in the test data. Element testing on rubber pad suspension elements and hydraulic shock absorbers (Section 2.2) were used to more adequately characterize viscous damping elements and supplement the truck test data.

2.1.3 Truck Test Articles

Truck test articles were chosen by the FRA to represent a cross section of trucks commonly used by U.S. railroads. From 1976 through 1980, eight trucks were tested: six three-axle designs and two two-axle designs. Table 2-4 lists the trucks tested and references the detailed test reports published for each truck. The following paragraphs describe the various truck designs, highlighting suspension system details.



a) Linear Spring



b) Bilinear Spring

Figure 2-12 Load-Deflection Characteristics for Linear and Bilinear Springs

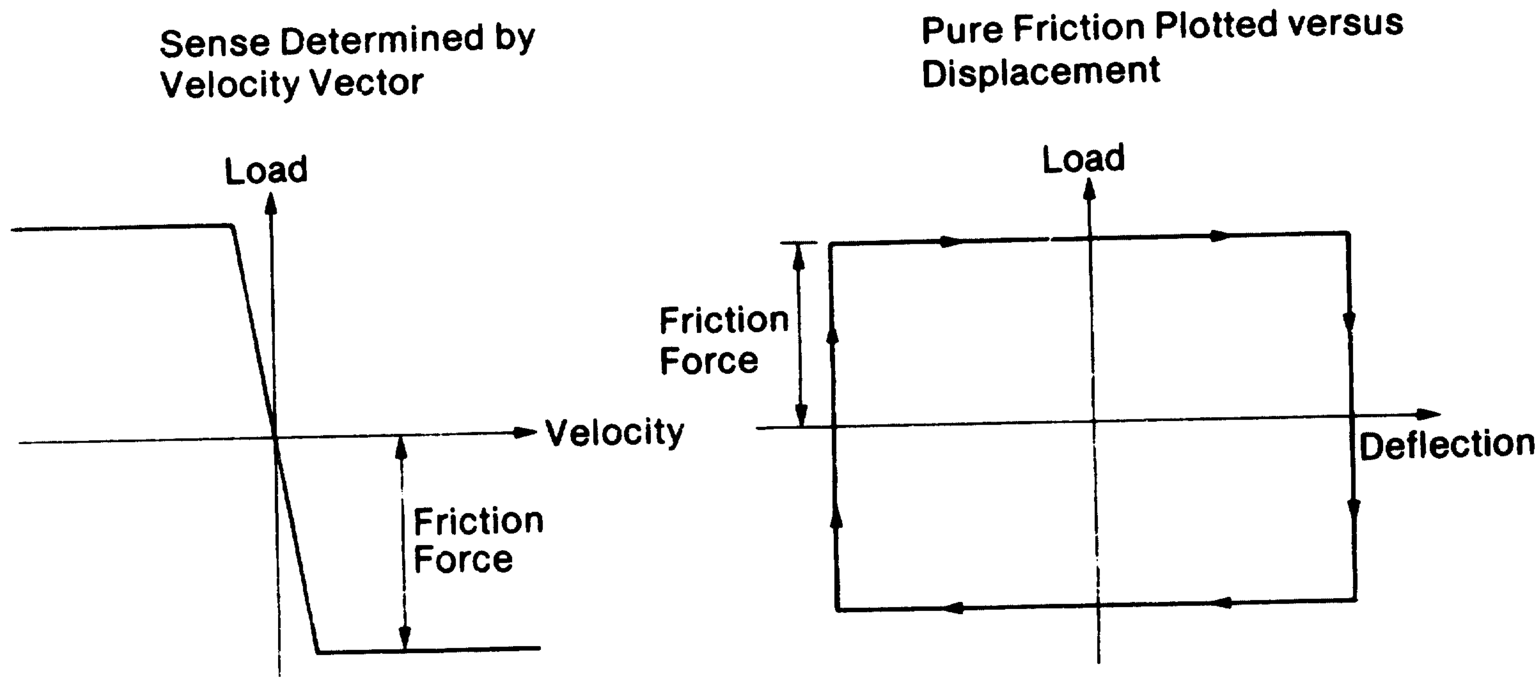


Figure 2-13 Hysterisis Characteristic of Pure Friction

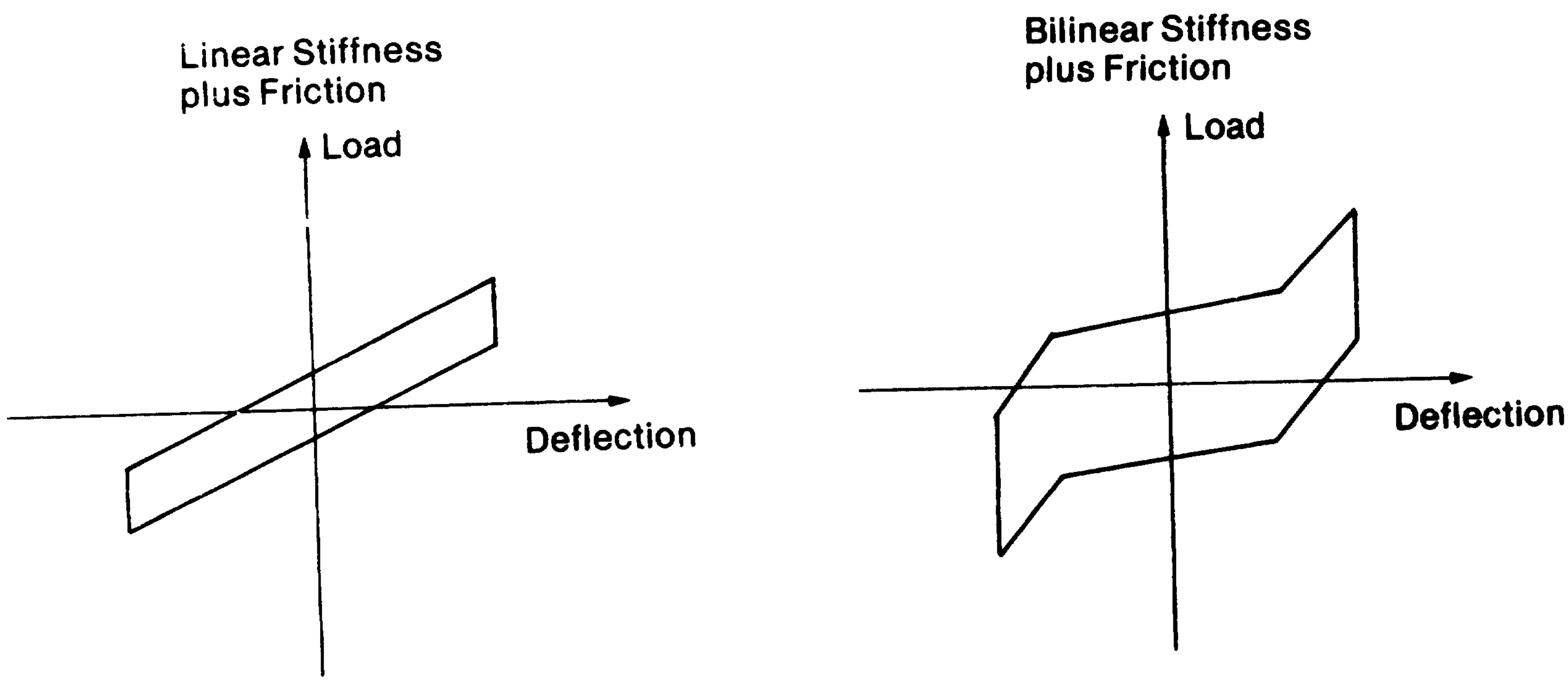


Figure 2-14 Hysterisis Characteristics of Combined Stiffness and Friction

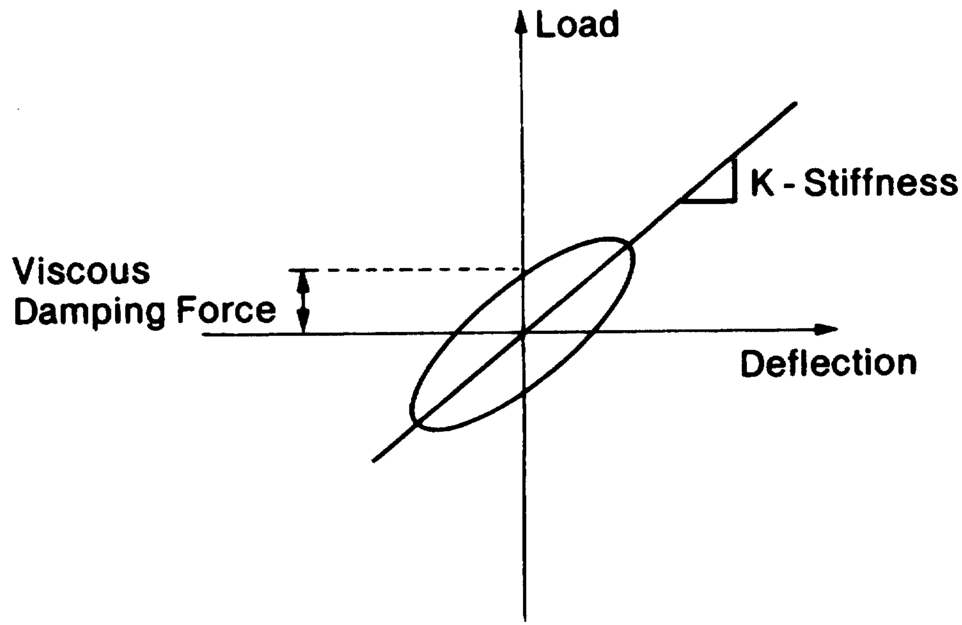


Figure 2-15 Hysterisis Characteristic of Combined Linear Stiffness and Viscous Damping

Table 2-4 Locomotive Truck Test Articles

TRUCK DESIGNATION	MANUFACTURER	DATE TESTING COMPLETED	TEST REPORT
3-AXLE			
HT-C (Hard Rubber)	EMD	October 1976	Reference 2
Flexicoil	EMD	January 1977	Reference 3
U30C	GE	March 1977	Reference 4
E60	GE	July 1977	Reference 5
HT-C (Soft Rubber)*	EMD	February 1978	Reference 6
E8*	EMD	October 1978	Reference 7
2-AXLE			
GPSS*	EMD	August 1979	Reference 8
GPSS (Inclined Rubber Pads)*	EMD	March 1980	Reference 8

*Tested at Ambient ($\approx 70^{\circ}F$) and Cold ($\approx 0^{\circ}F$) Conditions.

E8: An early vintage (1940's) three-axle EMD truck. The truck has a swing hanger secondary lateral suspension, with transverse leaf springs for the secondary vertical suspension. The primary vertical suspension has load equilization struts, which attempt to equalize vertical axle loads. No external dampers are employed on the truck. Figure 2-16 is a photograph of the E8 truck tested.

Flexicoil: An EMD three-axle design produced for SD type locomotives, prior to development of the HT-C truck design. The truck employs a standard secondary suspension with coil springs as the active suspension element. Two hydraulic shock absorbers are used in the primary vertical suspension system. They are located on the center axle. Figure 2-17 is a photograph of the Flexicoil truck during testing.

HT-C: This three-axle EMD design appeared on the market in the early 1970s. HT-C stands for High Traction-three axle. This truck also employs a standard secondary suspension utilizing single segment rubber pad springs as the active suspension elements. Two hydraulic shock absorbers are used on the center axle of the primary vertical suspension system. Early operational derailment problems led to design modifications including: the use of softer rubber pad springs and the addition of a hydraulic shock absorber in the secondary lateral suspension. Both versions of the truck were tested. Figure 2-18 is a photograph of the first HT-C truck tested.

U30C: A three-axle GE truck produced for diesel electric service. This truck has a standard secondary suspension with five-segment rubber pad springs as the active suspension elements. Friction snubbers are used to provide primary vertical suspension damping. Early versions of this truck employed four snubbers (front and rear axles). Later versions use only two snubbers (on the center axle). Figure 2-19 shows the U30C truck during testing.

E60: A three-axle GE truck produced for all-electric Amtrak service. The basic design of this truck is essentially the same as the U30C truck, with the exception of external damping devices. This truck employs eight hydraulic shock absorbers: four in the primary vertical suspension (front and rear axles), two between the truck frame and locomotive carbody to damp yaw motion, and two in the secondary lateral suspension. The truck configuration is similar to the U30C design shown in Figure 2-19.

GPSS: A two-axle EMD truck. The truck has a swing hanger secondary lateral suspension, with rubber pad springs for the secondary vertical suspension. Two versions of this truck were tested: the original version having rubber pad springs in compression, and the modified design having inclined rubber pad springs. The inclined rubber pad springs result in a slightly softer vertical suspension. Two hydraulic shock absorbers are used in the primary vertical suspension system (diagonally opposite on the front and rear axles). Figure 2-20 is a photograph of the original version GPSS truck tested. The two segment rubber pad springs are visible in the photograph. Figure 2-21 is a closeup of the inclined rubber pad springs in the modified truck (the second GPSS truck tested).

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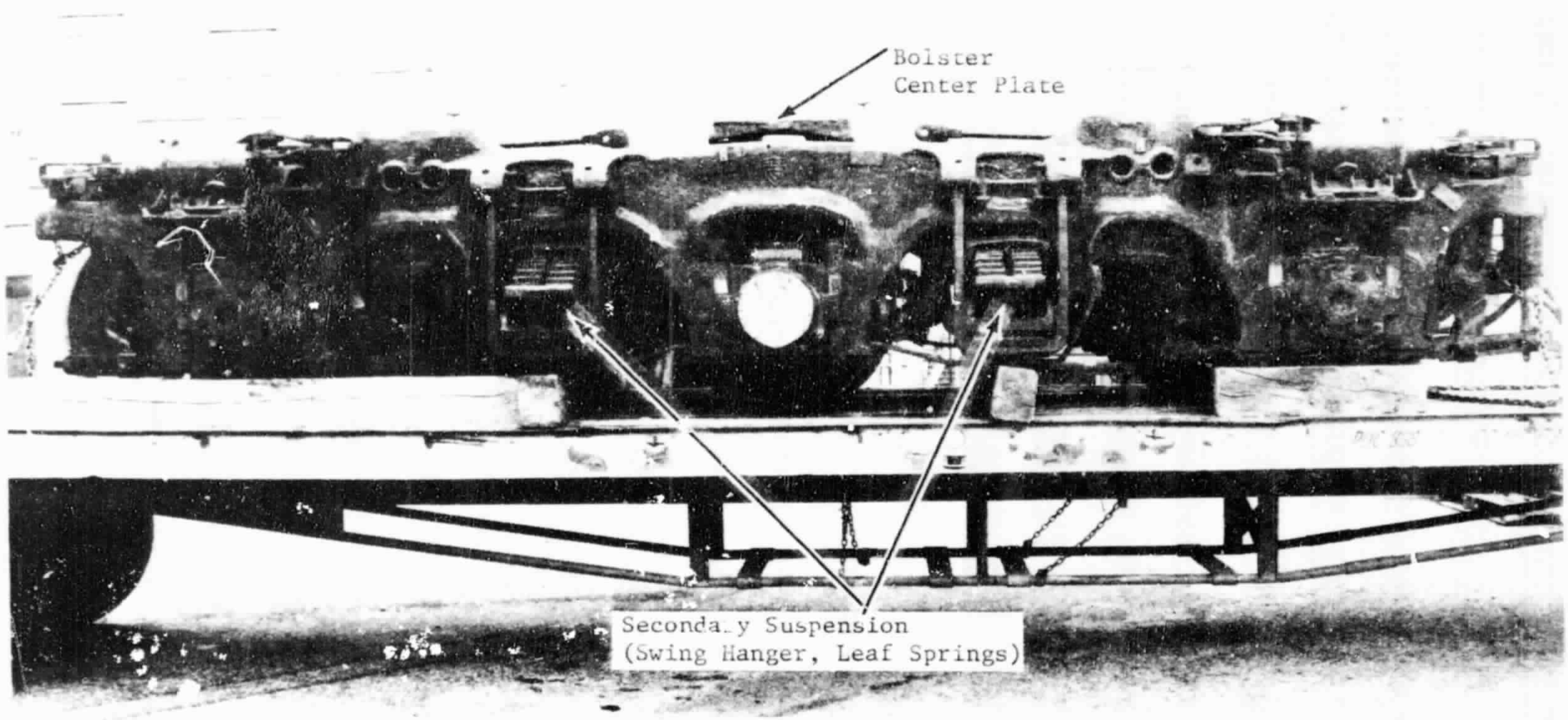

Figure 2-16 EMD E8 Truck

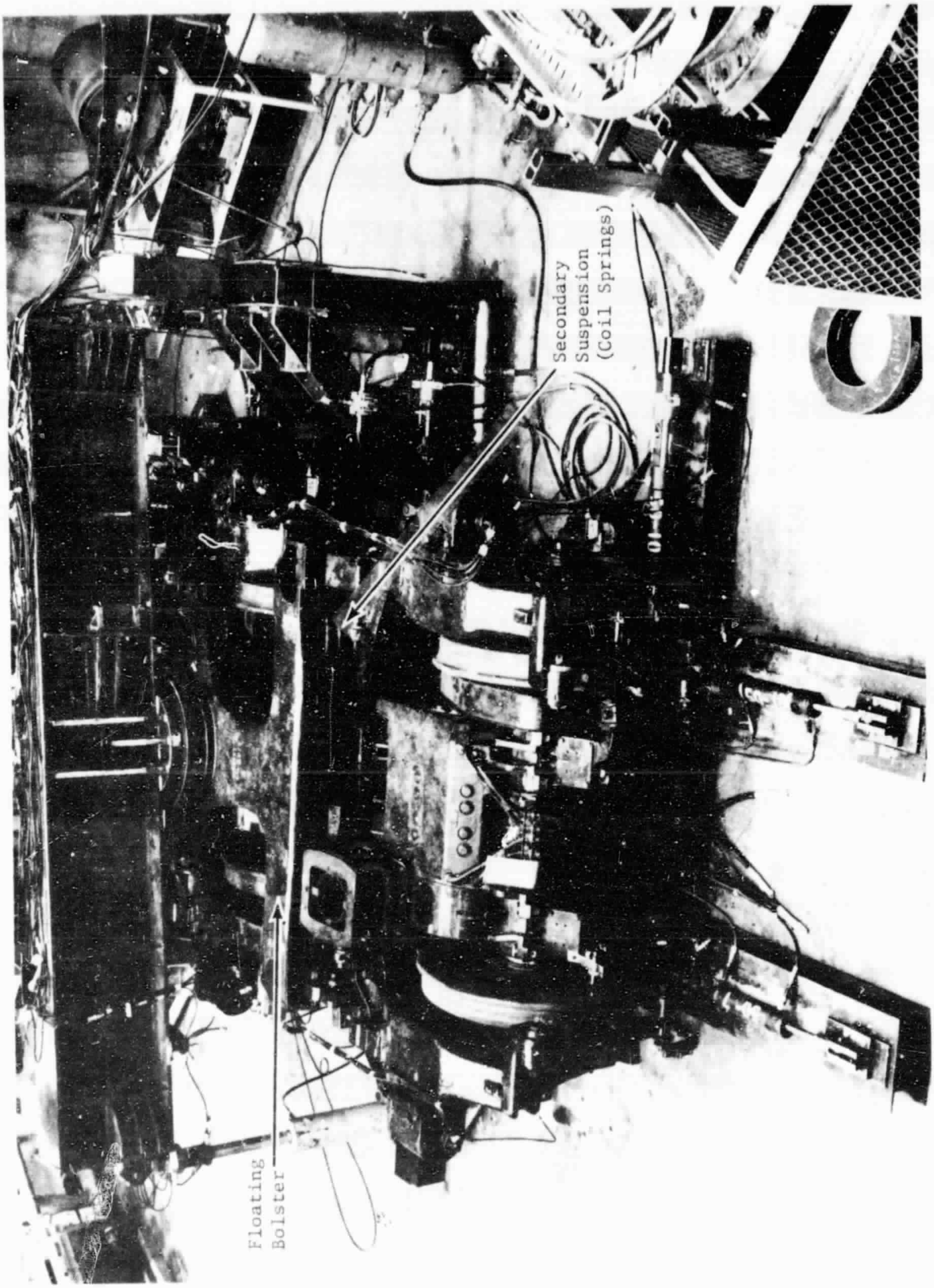

Figure 2-17 EMD Flexicoil Truck

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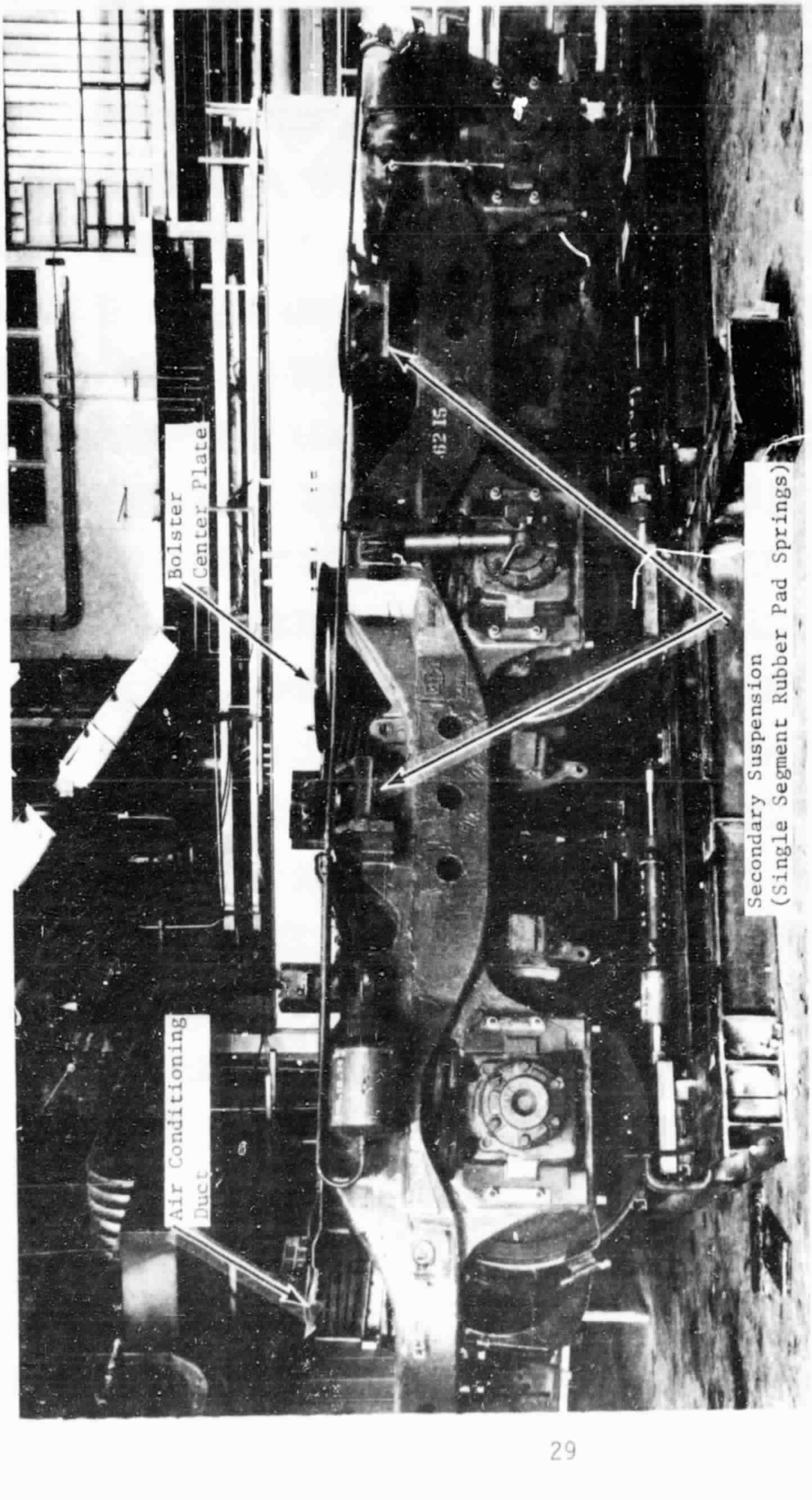

Figure 2-18 EMD HT-C Truck

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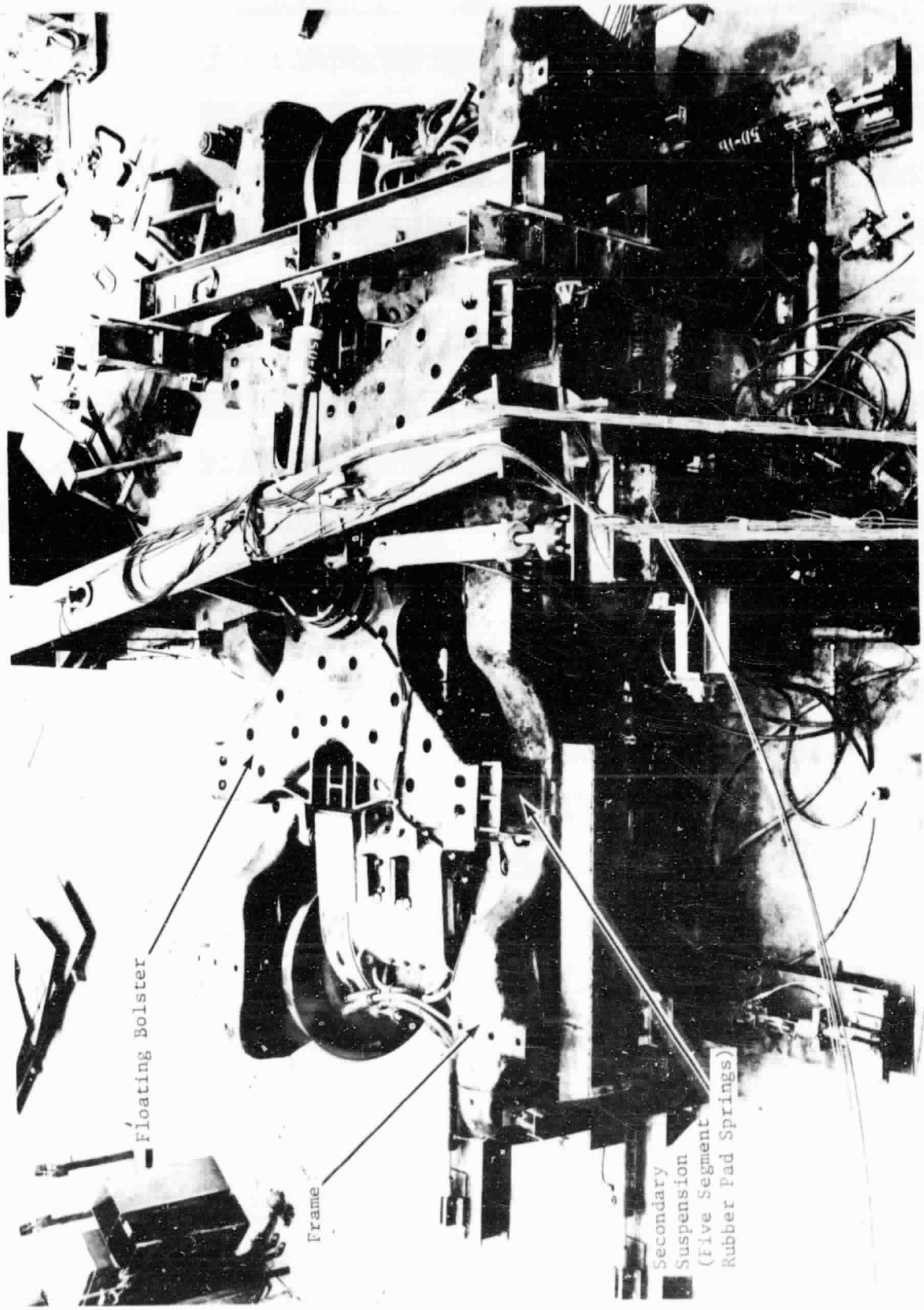

Figure 2-19 GE U30C Truck

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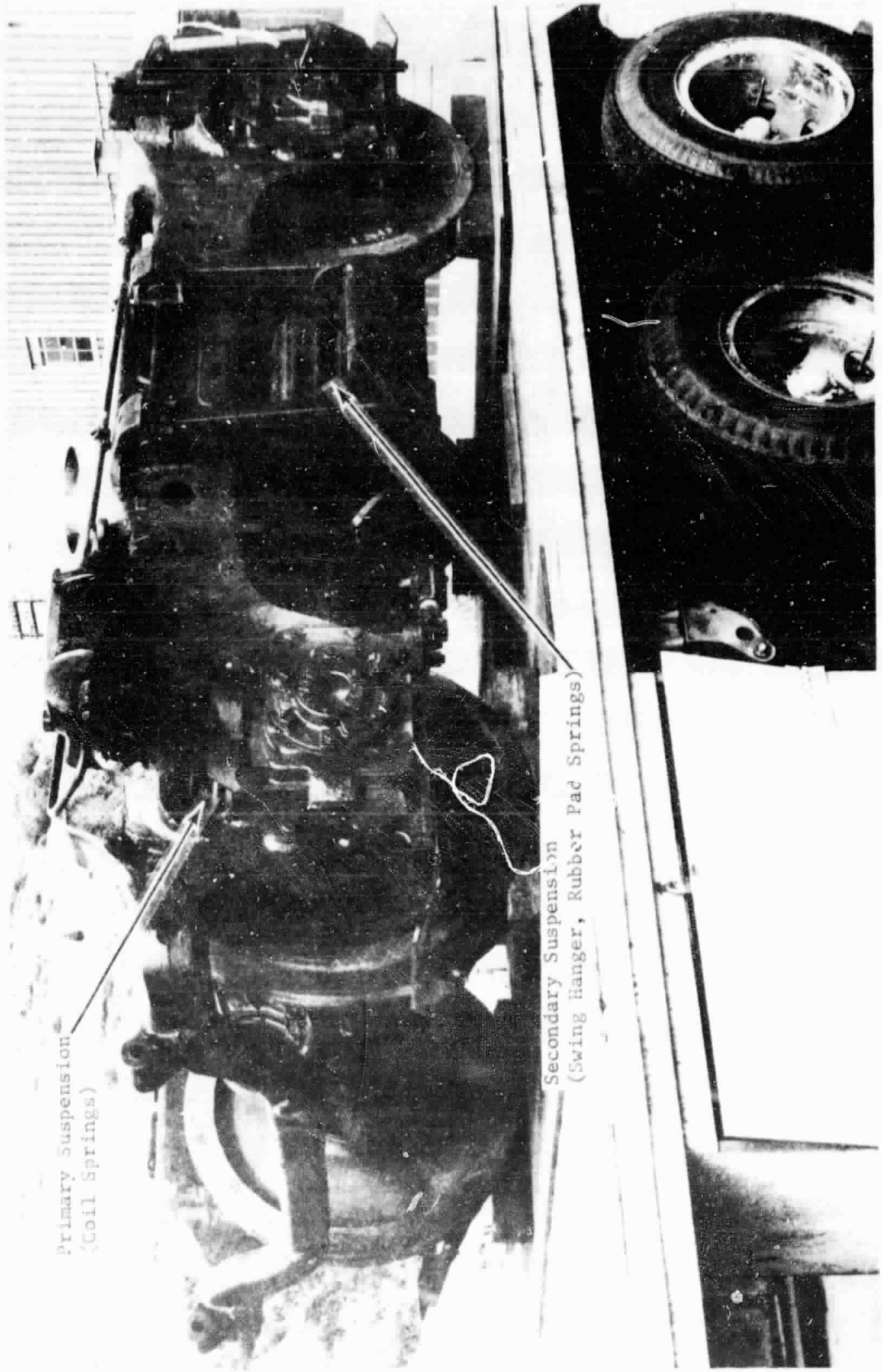

Figure 2-20 EMD GPSS Truck

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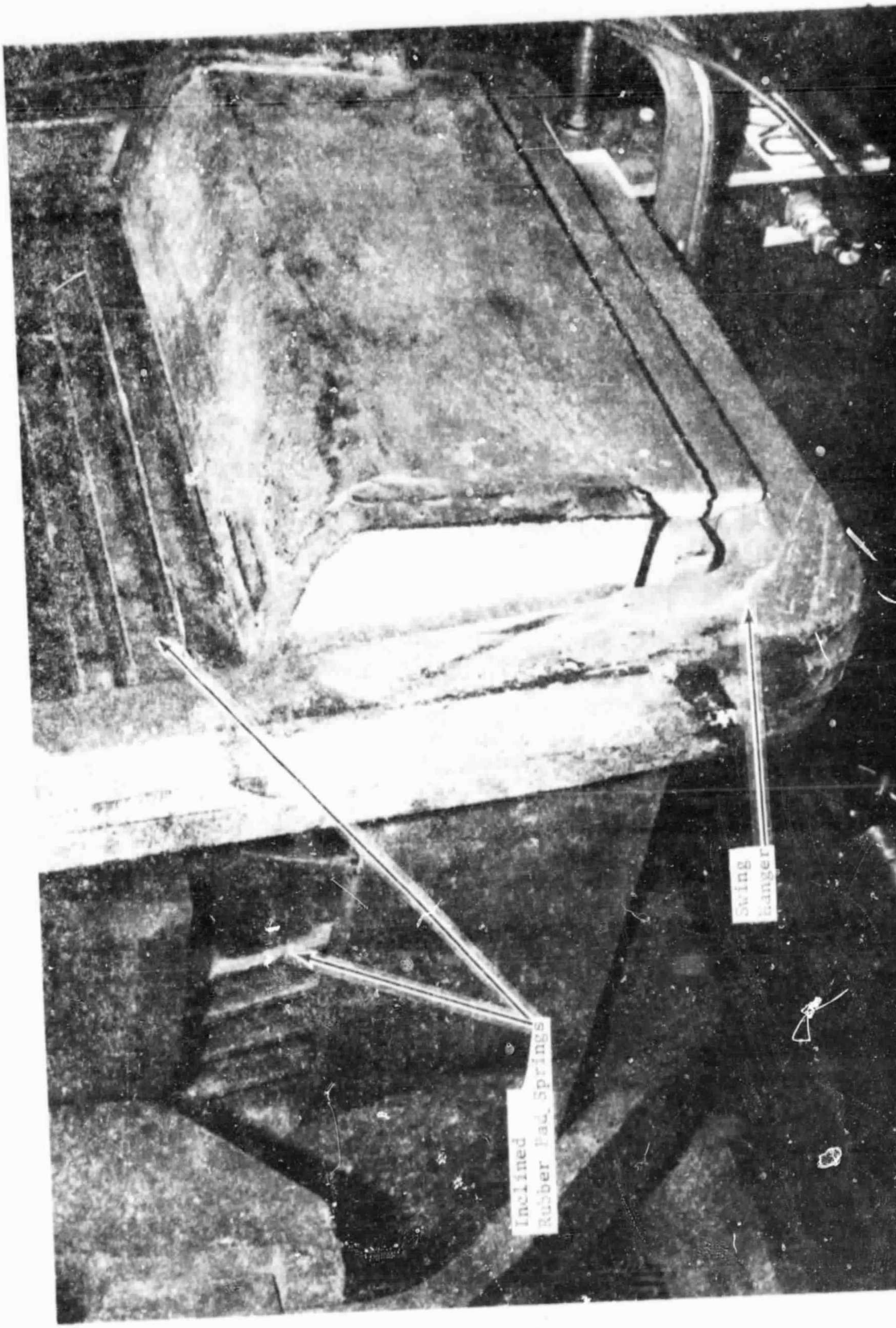


Figure 2-21 Inclined Rubber Pad Springs on Modified GPSS Truck

2.1.4 Results Summary

The test data acquired for each truck and a detailed description of the truck configurations are presented in References 2 through 8. The Appendix of this report contains a summary of the test data, along with other information needed for dynamic modeling. The references should be consulted for additional details. All trucks were tested with external damping devices removed.

Many of the trucks tested are available with a range of suspension stiffnesses to fit the particular service application. The measured values, presented in this report, and the detail test reports only reflect the particular trucks tested. In addition, state of wear can affect some of the measured data. Any analyses, using the data obtained during testing, should include consideration of sensitivity of analysis results to variations in suspension parameters.

Figures 2-22 and 2-23 show a comparison of the vertical and lateral stiffness of the eight trucks tested. Figure 2-22 shows the truck's equivalent vertical stiffness considering the secondary and primary suspension systems in series. Figure 2-23 is a comparison of the secondary lateral stiffnesses. Note that the swing hanger suspension used in the E8 and GPSS trucks significantly reduces the lateral suspension stiffness.

2.2 Component Test Program

In addition to the locomotive truck tests, three component test programs were conducted under the contract. The objective of these test programs was to investigate, in more depth, the characteristics of key suspension components. The following paragraphs summarize the three component test programs conducted.

2.2.1 Rubber Suspension Pad Element Test

The truck tests did not provide sufficient resolution to allow a complete understanding of the properties of the elastomeric pads used in truck secondary suspensions. Since the pad is a key element in determining truck behavior, a separate element test was performed.

A simple test fixture was constructed to allow measurement of rubber pad stiffness and damping, as a function of pad temperature and frequency of excitation. Figure 2-24 shows a sketch of the test setup. Two rubber pad springs were preloaded in series with approximately one-eighth the weight of a carbody, and a hydraulic actuator was used to load the pads in shear. Load-deflection characteristics were measured over a temperature range of -55 to 70°F and an excitation frequency range of 0.25 to 3.0 Hz.

Rubber pad springs from three trucks were tested: HT-C (hard rubber), HT-C (soft rubber), and E60. Pad stiffness and equivalent viscous damping coefficient were derived from load-deflection hysteresis characteristics measured during testing. Figure 2-25 shows the variation in pad stiffness, as pad temperature was lowered, for an excitation frequency of 0.25 Hz.

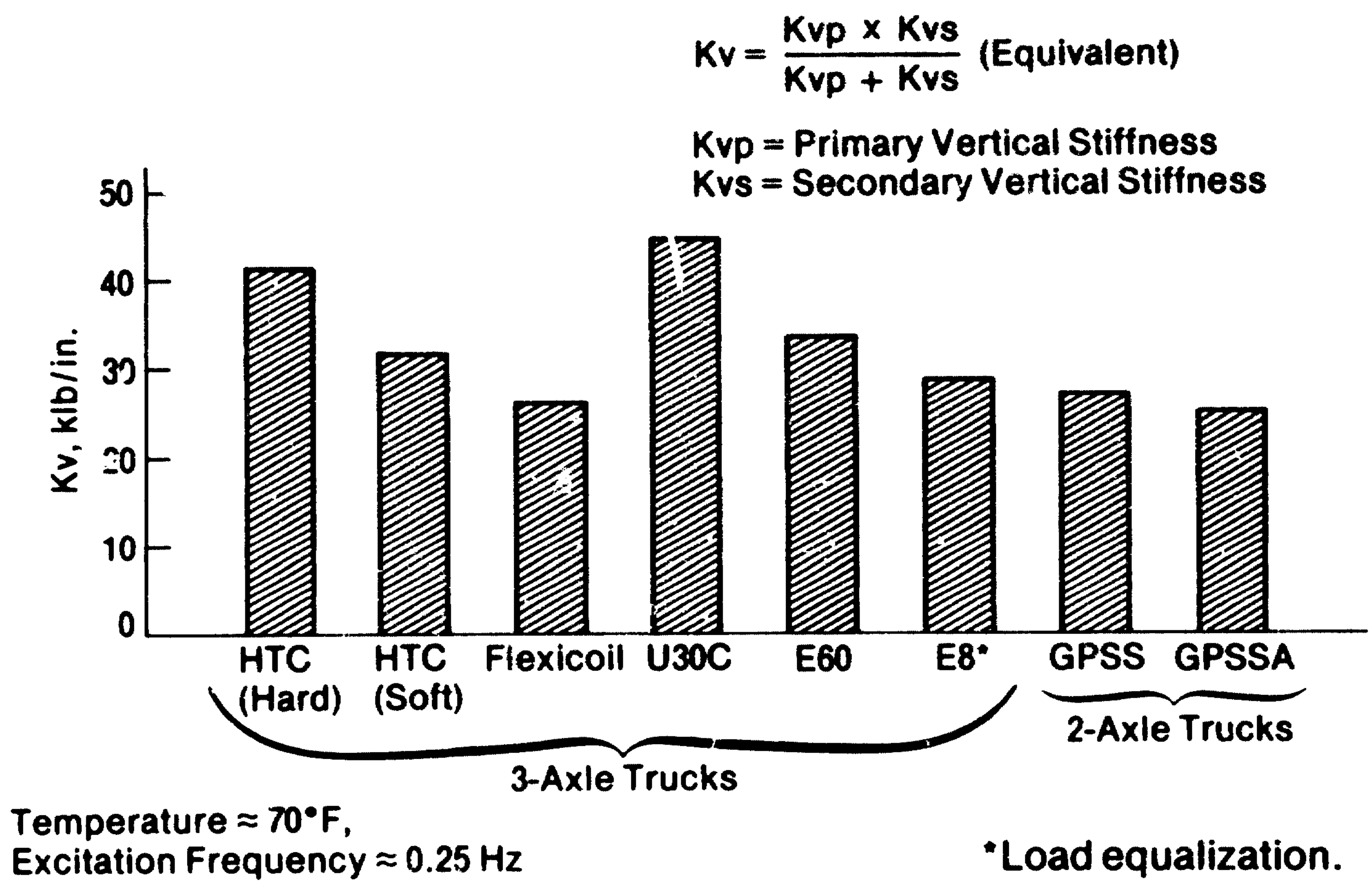


Figure 2-22 Truck Test Data: Comparison of Vertical Suspension Stiffness

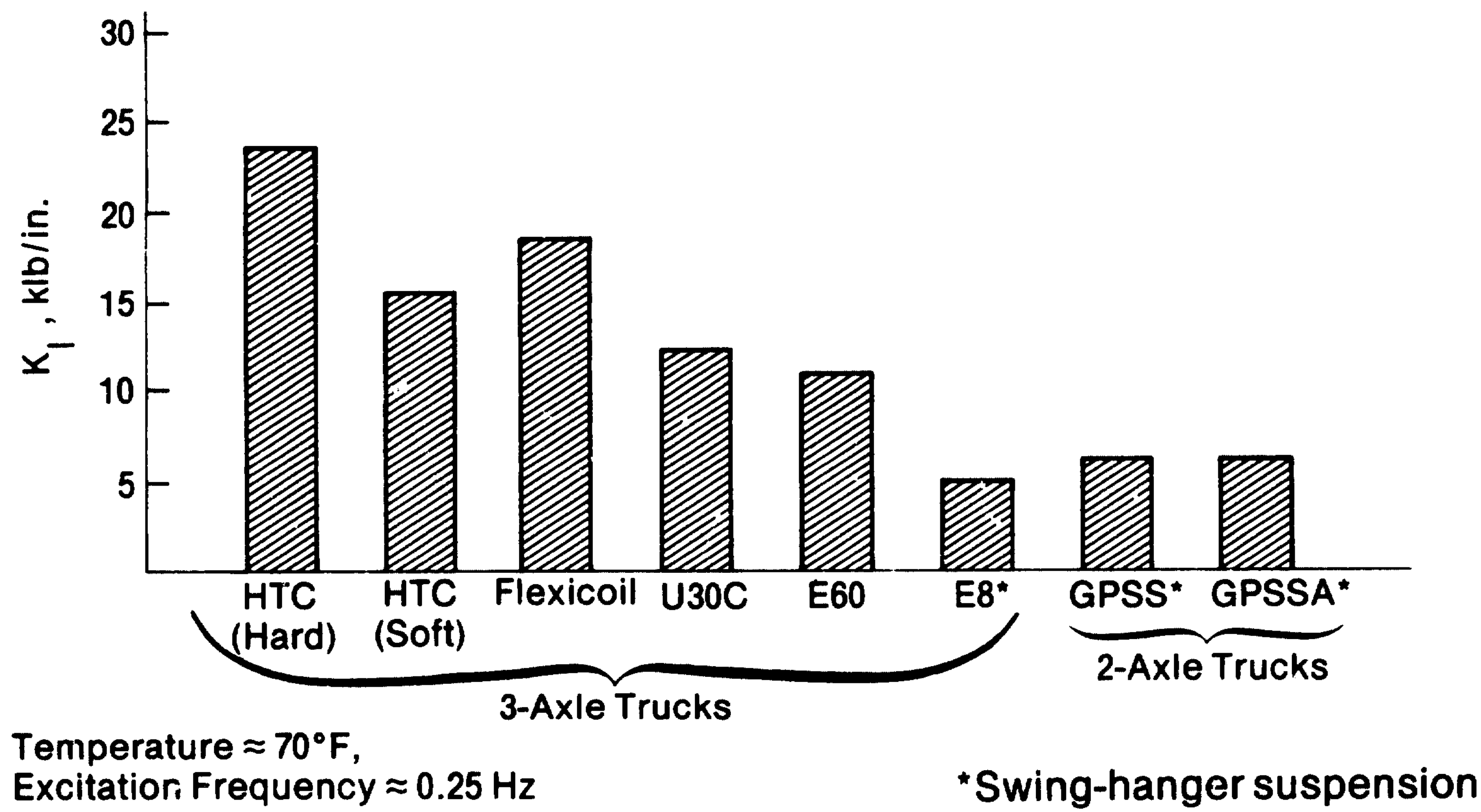
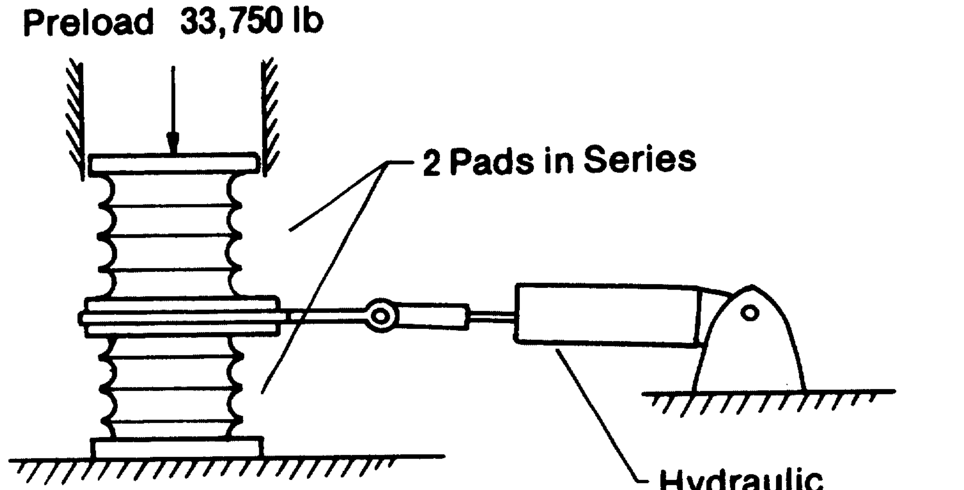


Figure 2-23 Truck Test Data: Comparison of Secondary Lateral Suspension Stiffness

• Secondary Suspension Pads Tested: HTC (Hard & Soft), E60



Excitation Frequency Range: 0.25 to 3.0 Hz
Temperature: -55 to +70°F

Figure 2-24 Setup for Rubber Suspension Pad Element Test

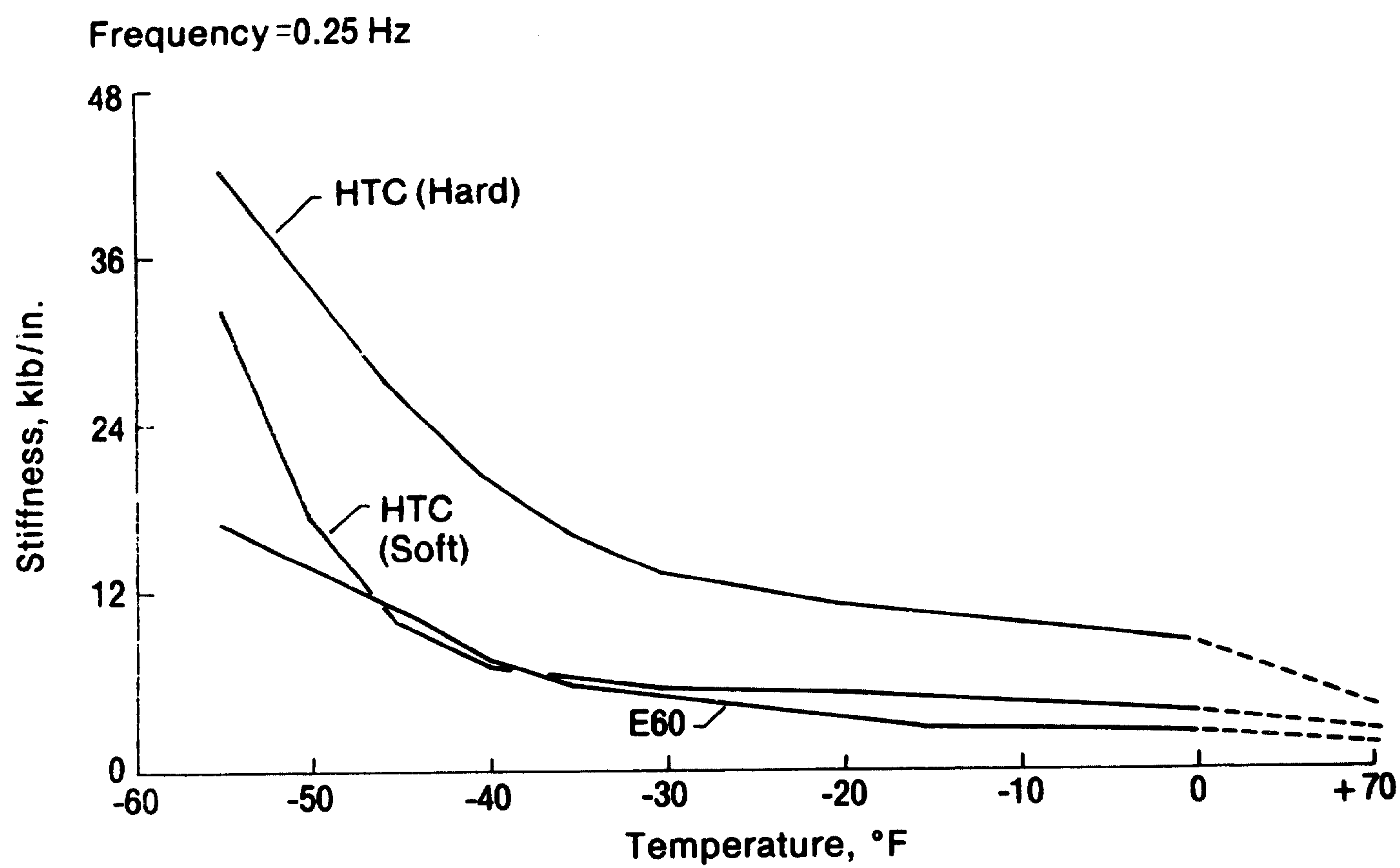


Figure 2-25 Rubber Pad Stiffness Versus Temperature

Figure 2-26 shows the increase in pad damping with a decrease in pad temperature. This trend is typical of polymer materials. In general, rubber pad springs are not good dampers, because they are functioning in the "rubbery region" as opposed to the "viscoelastic region". Polymer materials have a critical temperature called the glass transition temperature. At this temperature, the material enters the "plastic region" and is more susceptible to fracture damage. The glass transition temperature occurs below the viscoelastic region. For the pads tested, the viscoelastic region appears to be below -40°F. In service, the pads will be at a much higher temperature, due to internal heat generation.

Figure 2-27 shows the nonlinearity of pad stiffness, as excitation frequency is varied. Figure 2-28 shows the variation in damping coefficient, as excitation frequency is varied. Rubber pad characteristics can be analytically modeled with a variety of spring-damper analogs. The selection of a modeling method should depend on the analysis objectives. For many applications, a linear spring and damper in parallel may be sufficient.

2.2.2 Hyatt Bearing Lateral Bumper Element Test

Hyatt bearings (straight roller), in contrast with tapered Timken bearings, do not carry lateral thrust loads (along the axle shaft), except through rolling friction. Once the free play clearance is exceeded, a rubber bumper transmits the lateral load to the journal box. Figure 2-29 shows details of the bearing construction and identifies the location of the rubber bumper. Since bearing friction, resulting from rolling wheels, could not be simulated during truck testing (test friction was much higher), bumper stiffness could not be characterized. Consequently, an element test was conducted to measure the bumper's properties.

Testing was conducted on an MTS testing machine that plotted load versus deflection. During testing, the bumper was configured with its spacer and retainer similar to its installed configuration in the bearing to provide realistic boundary conditions. Figure 2-30 is a photograph of the Hyatt bearing bumper components.

Using the test data, a polynomial expression was developed relating load to bumper deflection.

$$\text{LOAD} = 1.440 \times 10^4\ \delta - 1.495 \times 10^5\ \delta^3 + 1.704 \times 10^7\ \delta^5$$

$$- 3.077 \times 10^8\ \delta^7 + 1.751 \times 10^9\ \delta^9, \text{ (lb)}$$

where

δ = bumper deflection (IN.)

2.2.3 Shock Absorber - Friction Snubber Element Test

A component test program was conducted to evaluate the damping characteristics of two representative external damping devices: a Delco 22012514 hydraulic shock absorber commonly used on EMD HT-C locomotive trucks, and a Houdaille 709702-11 friction snubber commonly used on GE U30C trucks. Figure 2-31 is a photograph of the test setup. An "oil derrick" fixture was

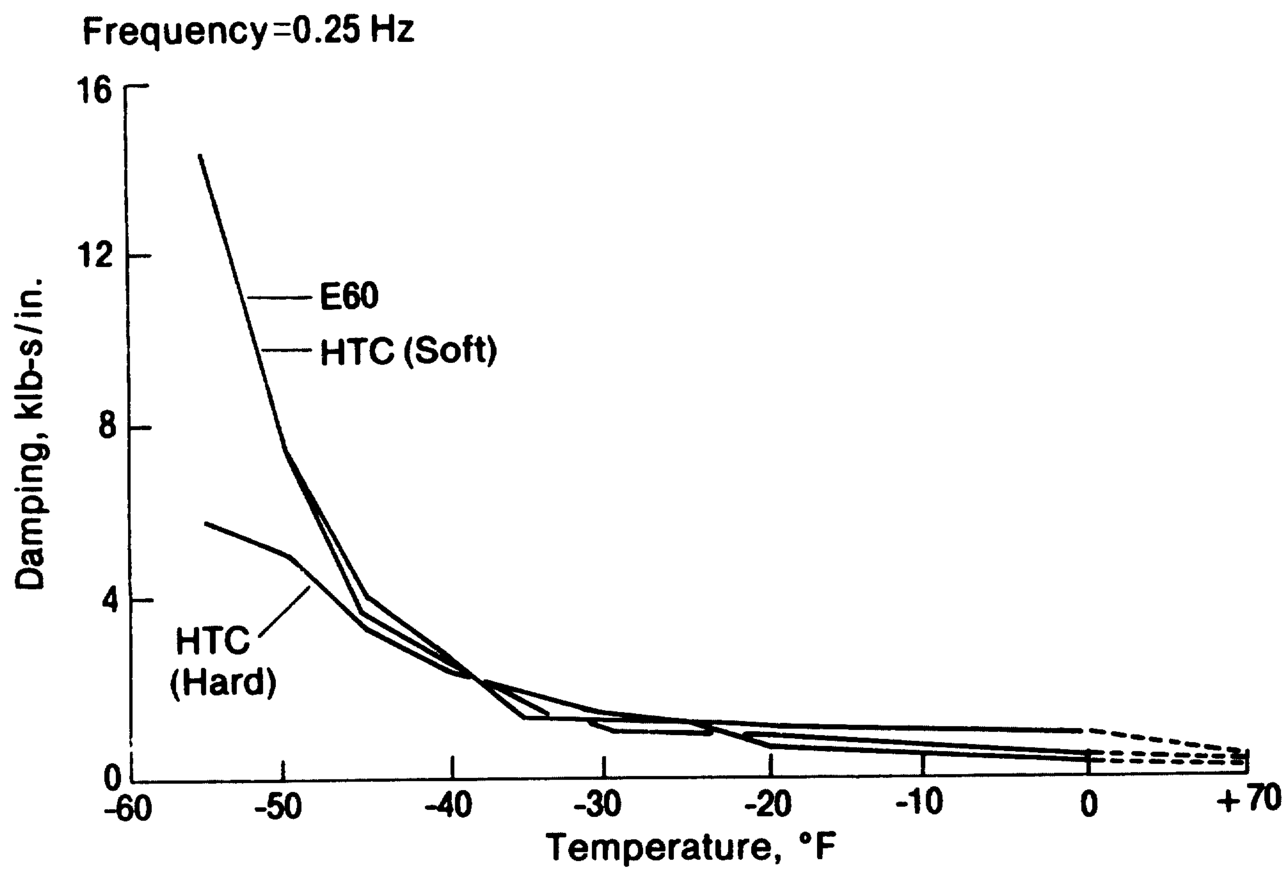


Figure 2-26 Rubber Pad Damping Versus Temperature

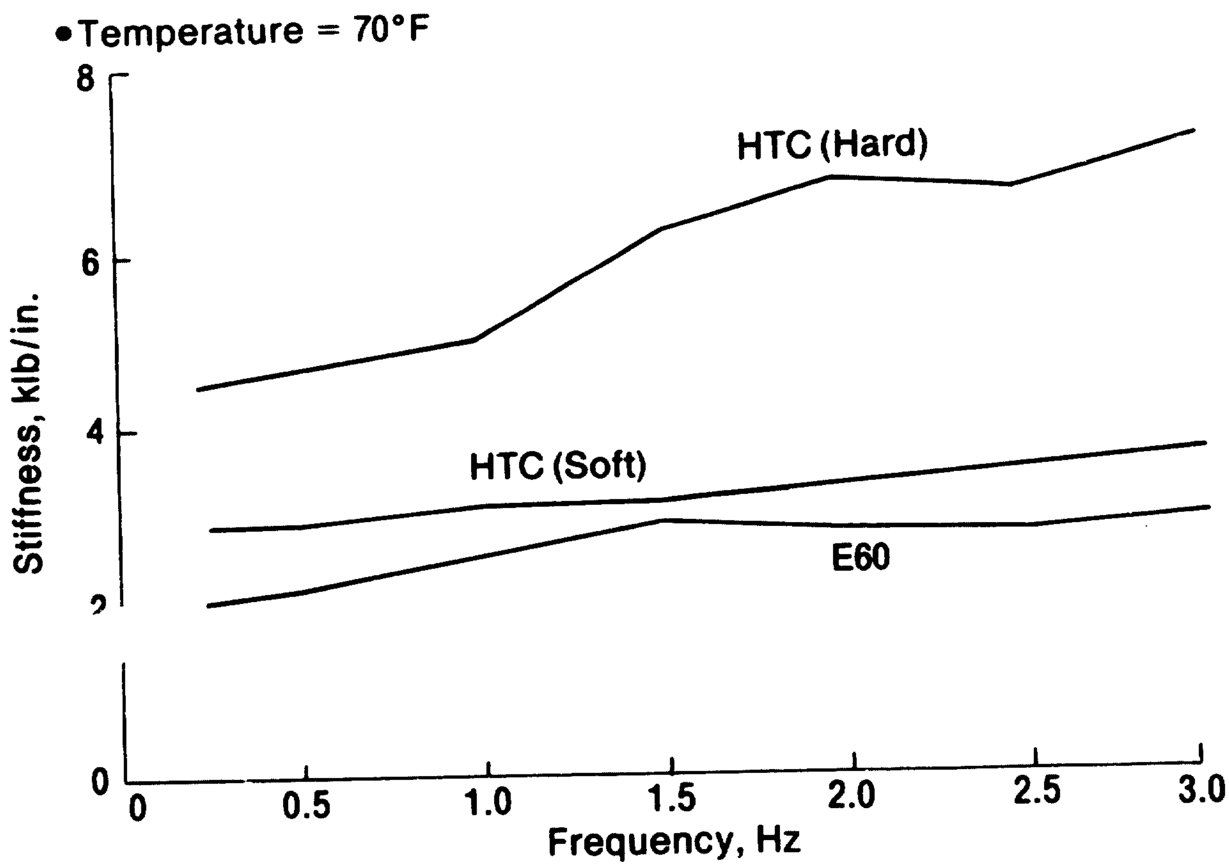


Figure 2-27 Rubber Pad Stiffness Versus Frequency

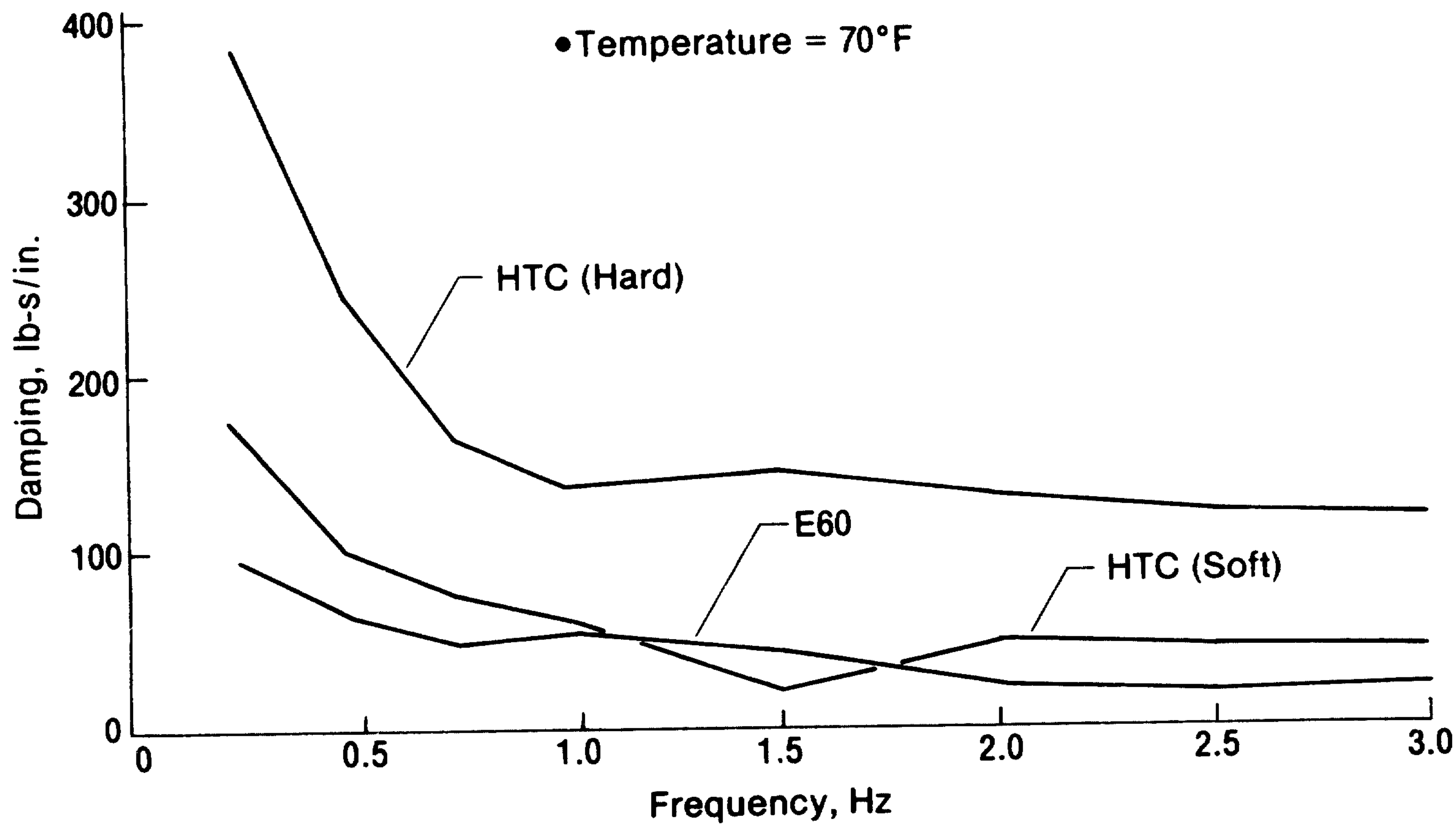


Figure 2-28 Rubber Pad Damping Versus Frequency

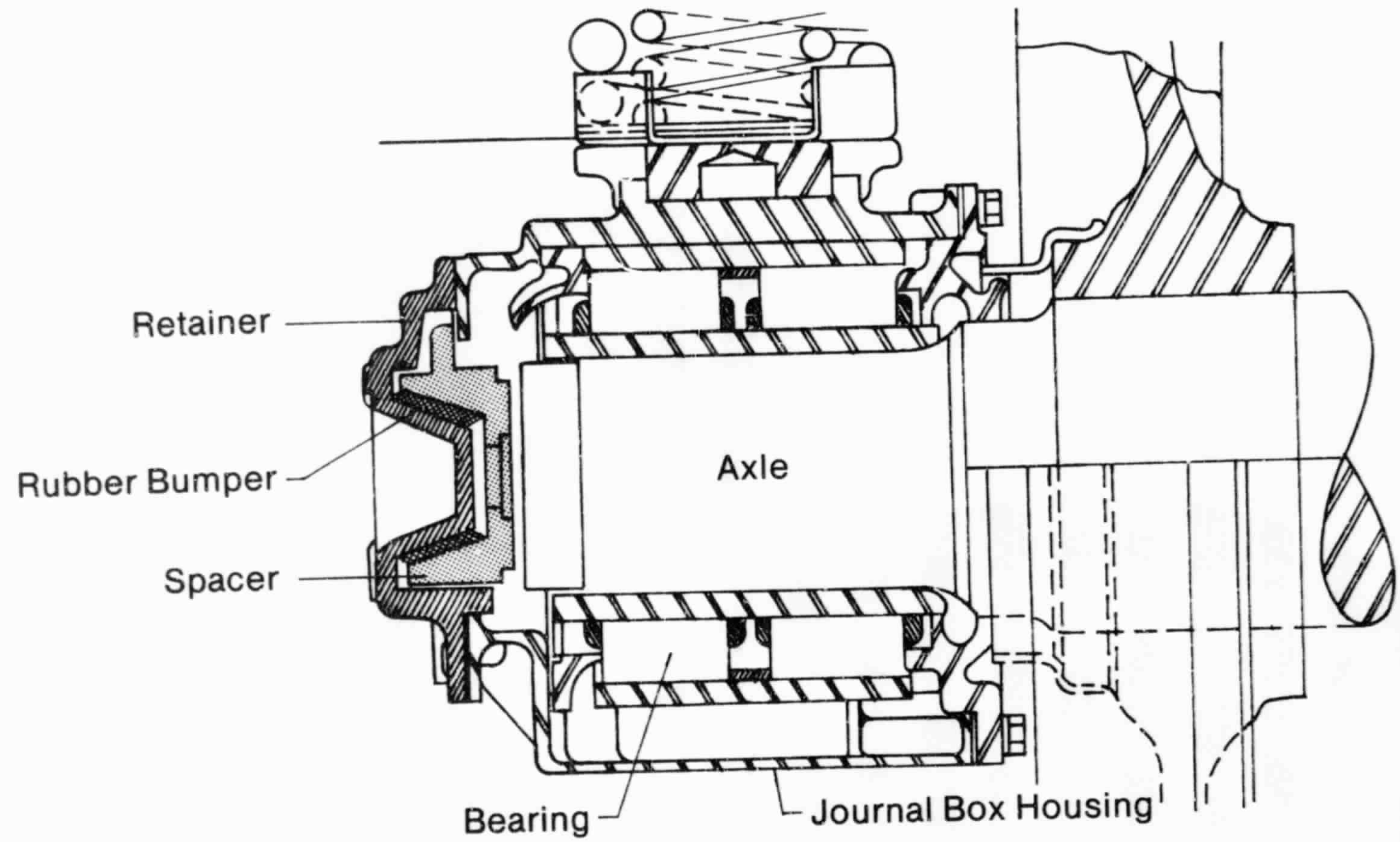


Figure 2-29 Hyatt Bearing Details

Figure 2-30 Hyatt Bearing Bumper Components

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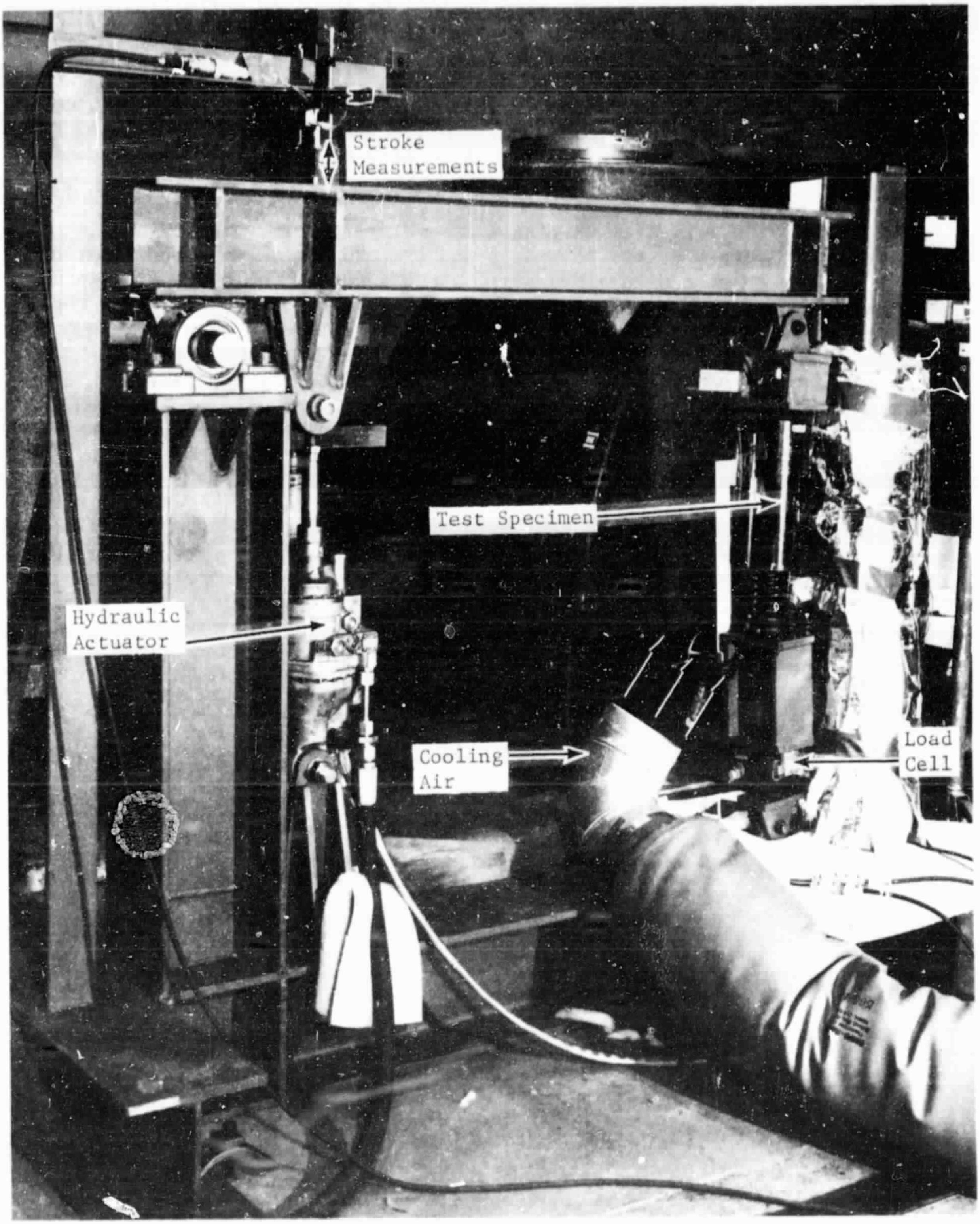

Figure 2-31 Shock Absorber Test Fixture

fabricated to facilitate a range of stroke amplitudes. A hydraulic actuator was used to load the test specimen utilizing the fixture's mechanical advantage to magnify the actuator's limited stroke. A strain gage load cell and two displacement transducers were used to measure the load deflection characteristics of the test specimens. A range of stroke velocities and displacements was obtained by varying excitation frequency and load. Specimens were tested over a temperature range from -50 to 165°F. Thermocouples mounted on the specimens were used to determine test temperatures. Liquid nitrogen-cooled air was used to control the specimen temperatures. Test details are presented in Reference 9. A summary of the results is presented below.

Figure 2-32 is a plot of damping force versus stroke velocity showing a comparison between measured test data and the manufacturer's specification average for the Delco 22012514 hydraulic shock absorber (commonly used on the EMD HT-C truck). The shaded area encompasses the test data for the temperature range from -50 to 100°F. These data correlate well with the manufacturer's specification average. At higher temperatures, however, the shock absorber's performance falls off significantly. The boundary envelope of the measured data between 100 and 165°F is shown. No attempt has been made to determine the temperature of a shock under operational conditions; however, prolonged travel at low speeds over very rough track could result in higher than normal temperatures, due to stroking of the shock.

Figure 2-33 summarizes the test data for the Houdaille 709702-11 friction snubber. All measured data met or exceeded the manufacturer's specification range, even for high temperature tests. Comparing the characteristics of the friction snubber with those of the hydraulic shock absorber, the basic difference between a velocity-dependent damping device (the shock absorber) and a damping device based on friction (the snubber) is evident. The snubber provides a fairly constant damping force, regardless of velocity.

- Delco P/N 22012514 Hydraulic Shock Absorber
- Stroke: 0.25 to 2.0-in. DA
- Frequency: 0.25 to 4.0 Hz

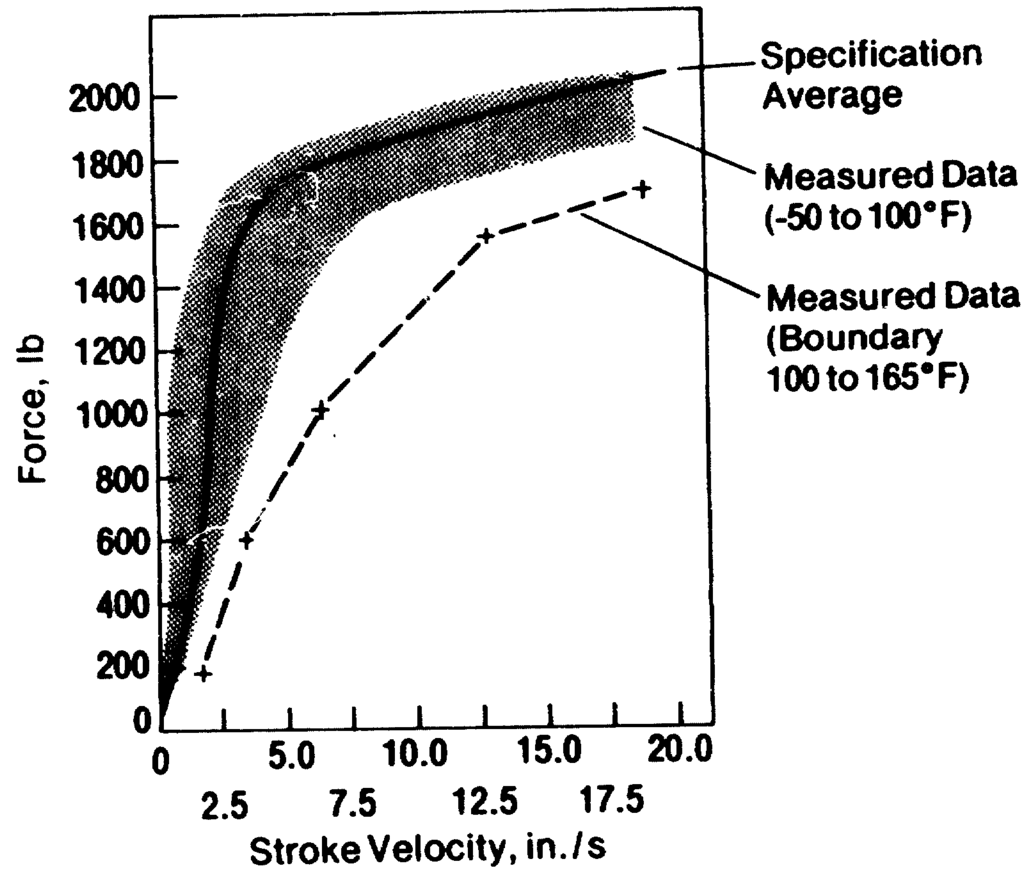

Figure 2-32 Element Test Data: HT-C Hydraulic Shock Absorber

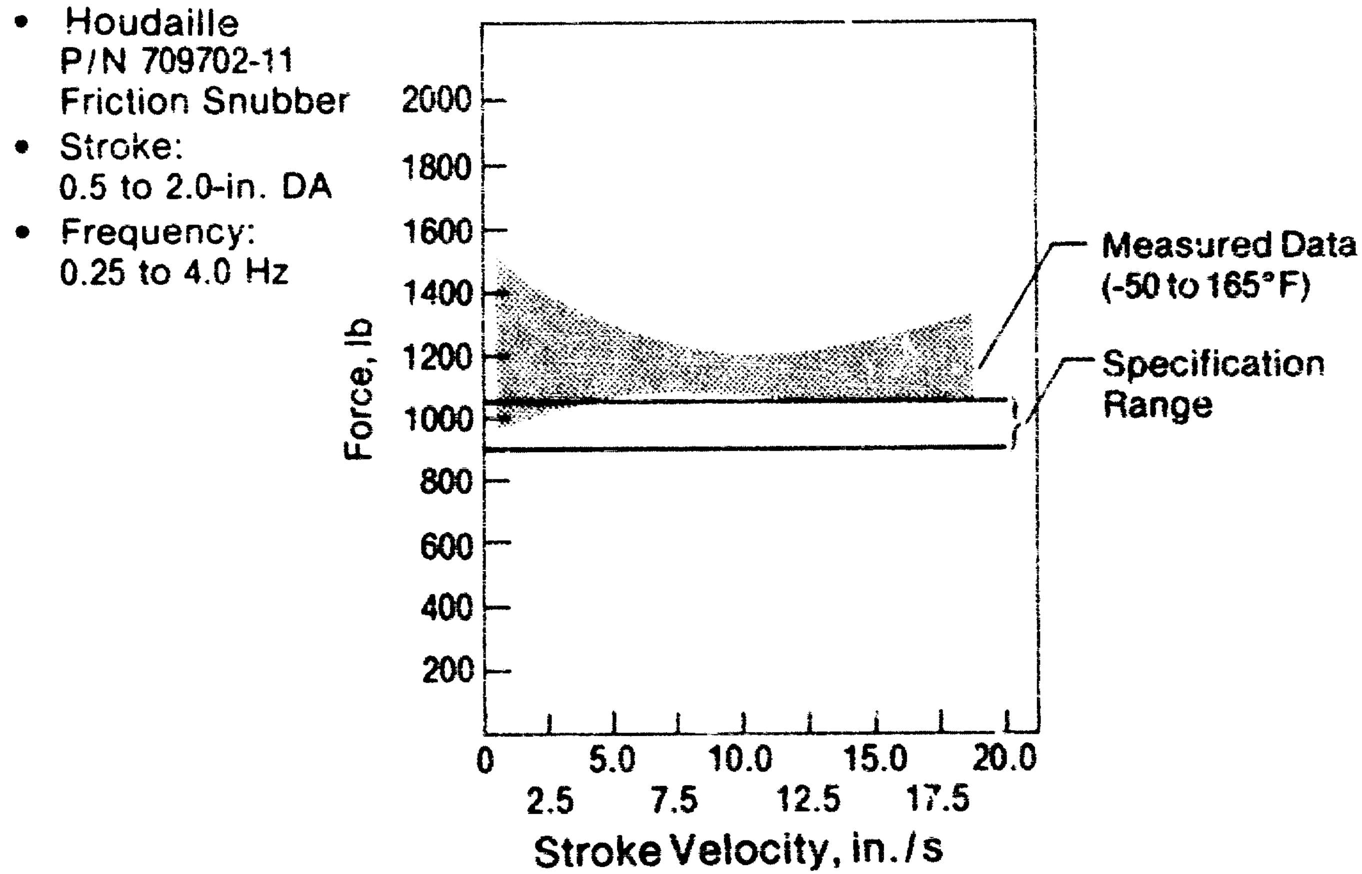

Figure 2-33 Element Test Data: U30C Friction Snubber

The testing methodology developed under this contract, and the data collected, have a variety of important applications to the railroad industry. One important application has been demonstrated by the development of a methodology for the derailment safety analysis of six-axle locomotives, documented in Reference 1. A computer code has been developed to simulate the nonlinear response of a locomotive to track geometry defects and calculate various measures of safety for evaluation. The code is user-oriented and provides the flexibility to modify the model and measures of safety, if so desired. Input data are hardware-oriented, and the truck data obtained in the test program form the basis for the truck test data bank. Figure 3-1 presents the approach used in developing the nonlinear locomotive simulation model.

The model simulates the nonlinear time response of a locomotive to track geometry defects on curved or tangent track. Track geometry defects may be specified in terms of vertical, cross-level, or gage perturbations. Complex defects can be constructed by superposition. Wheel/rail interactions are modeled in detail. The model includes a nonlinear creep formulation and also simulates wheel flanging. In addition, constant coupler forces and wind loads may be specified. Figure 3-2 shows the 15 degrees-of-freedom used to describe a locomotive in the model.

Two studies (documented in Reference 1) were performed to demonstrate applications of the model, useful to the railroad industry. A parameter sensitivity analysis was performed, in the first study, using the SDP40F locomotive. The objective was to determine the relative importance of various truck, track, and operational parameters to operational safety. Twenty-one different parameters were evaluated. Figure 3-3 shows an example sensitivity plot generated during this study. Shown is the computed sensitivity (in terms of lateral/vertical load ratios) to the limit force of the secondary lateral damper. The second study was a comparative analysis of three different locomotives subjected to the same set of rail geometry defects.

The testing techniques developed under this study allow the characterization of the locomotive truck as a system. Hence, complex interactions within the truck can be analyzed in the laboratory reducing the need for expensive field testing. The data obtained from the laboratory tests, combined with powerful analytical tools, such as the one discussed above (Reference 1), have general applicability in the design, maintenance, and operational evaluation of locomotives. Several potential applications are delineated below.

The methodology can be used as a predictive technique in:

1) The determination of maximum safe operating speeds for various classes of track based on locomotive dynamics;

2) The determination of critical track geometry defects;

3) The determination of minimum track strength requirements;

4) The development of appropriate locomotive maintenance standards;

5) The determination of derailment mechanisms;

6) Determination of the relative importance of truck, track, and operational parameters to operational safety;

7) The mechanical design of locomotive suspension components; and

8) The evaluation of new and/or modified locomotive designs, prior to introduction into service.

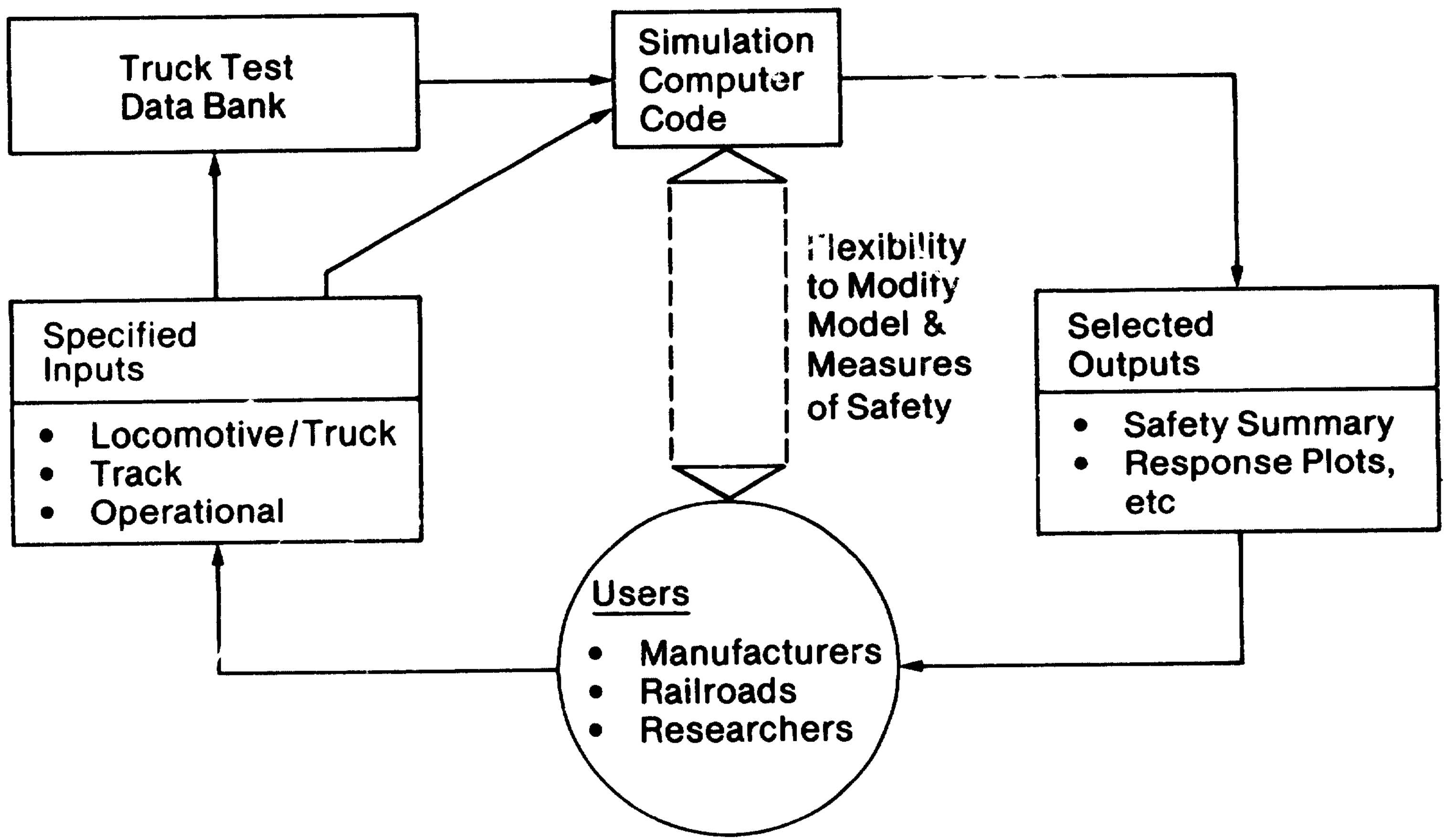

Figure 3-1 Nonlinear Locomotive Simulation, Modeling Approach

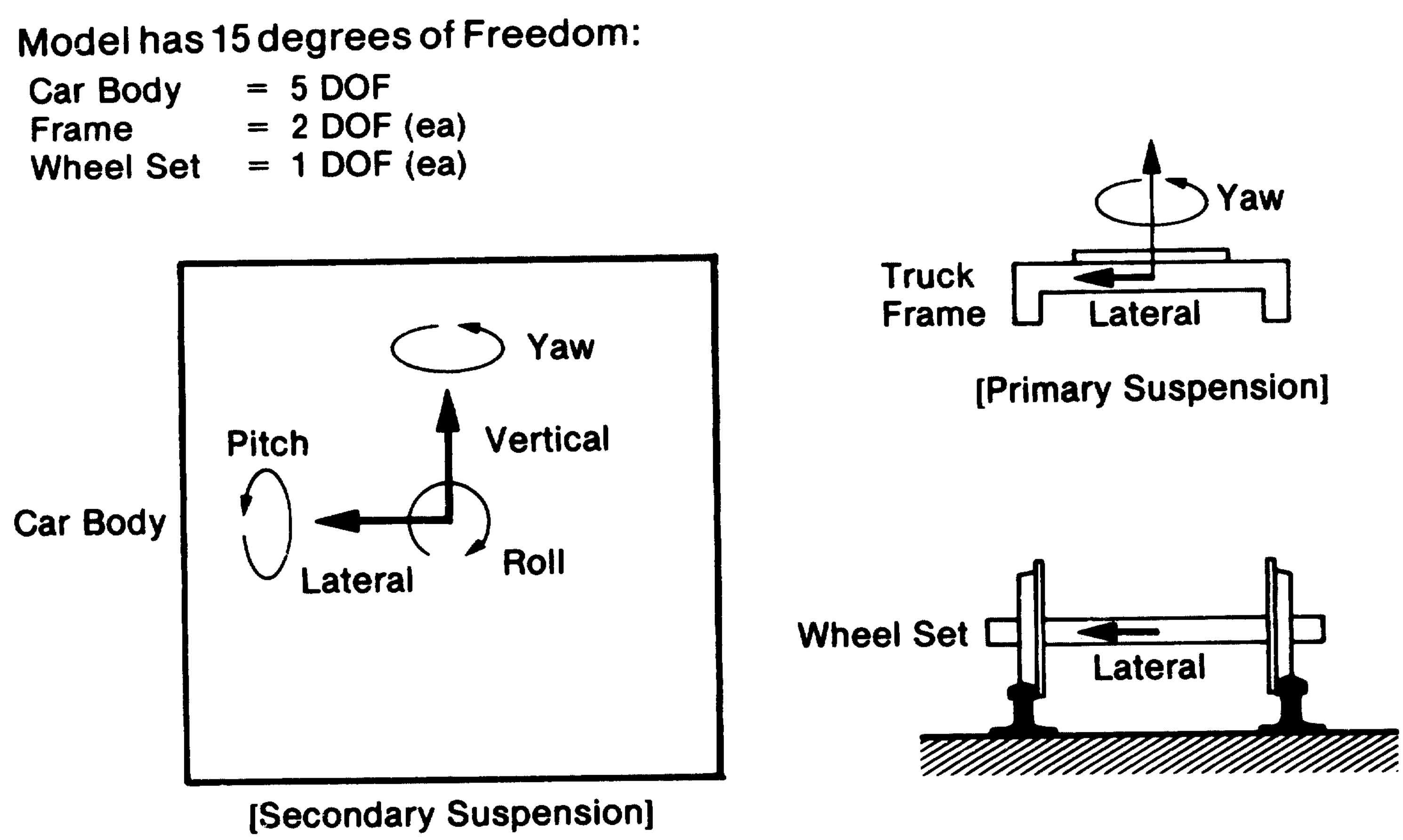

Figure 3-2 Nonlinear Locomotive Model Degrees of Freedom

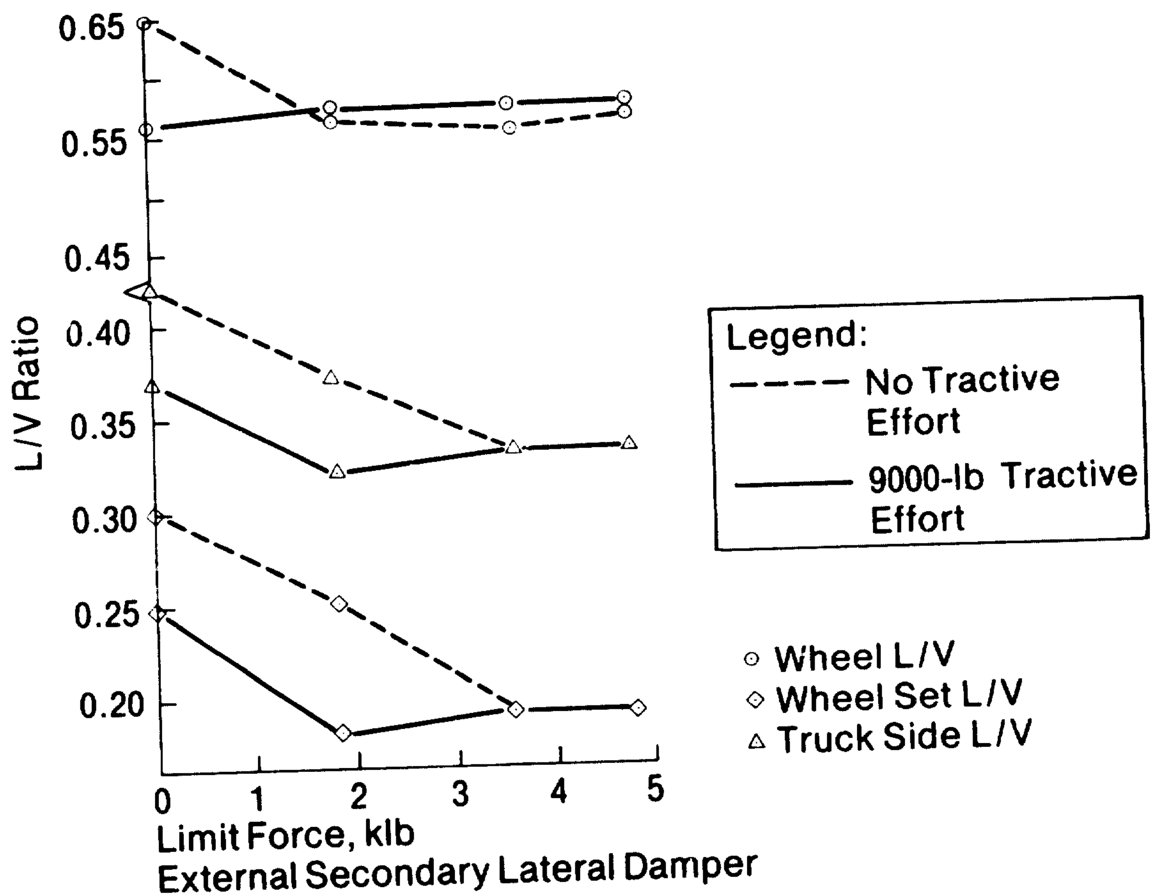

Figure 3-3 *Computed Sensitivity of L/V Ratios to Secondary Lateral Damper Limit Force; SDP40F Locomotive*

4.0 CONCLUSIONS AND RECOMMENDATIONS

This contract has led to the development and refinement of a methodology that allows the testing of a locomotive truck as a system. The testing methodology has been used to characterize eight locomotive trucks currently in service to form the core of an expandable truck data base.

An analytical tool that uses this data and simulates the nonlinear response of locomotives to track geometry defects has been developed. Through use of the test data and analytical tools, such as the one developed under this contract, many long range benefits to the railroad industry are possible including:

1) Reduced derailment risk;
2) Improved ride quality; and
3) Wear minimization.

The studies conducted under this contract have demonstrated the value to the industry of a complementary test and analysis program for evaluating locomotive/truck dynamic performance. The rudimentary test and analytical methodology are now available to facilitate the development of a centralized program for evaluating and optimizing truck designs. It is recommended that the knowledge gained under this contract be used as a source of ideas and guidelines for the development of a consolidated truck evaluation facility.

The establishment of a centralized facility, available to the entire industry, is attractive for several reasons:

1) It represents a neutral ground for performing standard-repeatable tests and comparing results;

2) It allows for a central repository of test data;

3) Personnel training can be minimized; and

4) It facilitates the collection, development, and enhancement of analytical tools for use by the industry.

Overview, policy guidance, advice and recommendations from the railroads, the railroad supply industry, the Railway Progress Institute, governmental agencies, and the Association of American Railroads are imperative for the successful development of such a facility.

Through the proposed facility, railroads would have the opportunity to evaluate the performance of prototype truck designs, prior to purchasing even limited quantities for road evaluation. The test/evaluation procedures would also allow identification of extreme conditions in a laboratory environment, minimizing real risks. Within the laboratory confines, fine tuning of designs could be accomplished and ultimately performance specifications for advanced truck designs could be developed. The proposed evaluation program will present the railroad industry with a new, cost effective tool for managing and enhancing locomotive fleet operations.

5.0 REFERENCES

1) P. P. Marcotte and K. J. R. Mathewson, "Methodology Development for the Derailment Safety Analysis of Six-Axle Locomotives", MCR-81-560, Contract NAS8-29882, Martin Marietta Corporation, May 1981.

2) P. W. Abbott, "Track/Train Dynamics Test Results: HT-C Truck Static Test", MCR-77-100, Contract NAS8-29882, Martin Marietta Corporation, 15 March 1977.

3) P. W. Abbott, "Track/Train Dynamics Test Results: Flexicoil Truck Static Test", MCR-77-211, Contract NAS8-29882, Martin Marietta Corporation, 15 June 1977.

4) P. W. Abbott, "Track/Train Dynamics Test Results: U30 Truck Static Test", MCR-77-240, Contract NAS8-29882, Martin Marietta Corporation, 5 November 1977.

5) P. W. Abbott, "Track/Train Dynamics Test Results: E60 Truck Static Test", MCR-77-297, Contract NAS8-29882, Martin Marietta Corporation, 10 October 1977.

6) P. W. Abbott, "Track/Train Dynamics Test Results: New HTC Truck Static Test", MCR-78-519, Contract NAS8-29882, Martin Marietta Corporation, 1 March 1979.

7) P. W. Abbott, "Track/Train Dynamics Test Results: EMD E-8 Truck Static Test", MCR-78-645, Contract NAS8-29882, Martin Marietta Corporation, July 1979.

8) Barbara A. Shelton and Robert L. Berry, "GPSS/GPSS (Modified) Two-Axle Locomotive Trucks: Static Test Results", MCR-80-565, Contract NAS8-29882, Martin Marietta Corporation, August 1980.

9) Barbara A. Shelton, "Shock Absorber/Friction Snubber Test Report", MCR-80-564, Contract NAS8-29882, Martin Marietta Corporation, July 1980.

10) "Car and Locomotive Cyclopedia of American Practices", First Edition 1966, Third Edition 1974, Fourth Edition 1980, Simmons-Boardman Publishing Corporation.

11) M. Coltman, R. Brantman, P. Tong, "A Description of the Tests Conducted and Data Obtained During the Perturbed Track Test", Report No. FRA/ORD-80/15, January 1980.

12) "Locomotive Truck Assemblies", Simmons-Boardman Publishing Corporation, 1981.

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Appendix
Truck Test Data Summary

A.1 EMD E8 Truck

A.2 EMD Flexicoil Truck

A.3 EMD HT-C Truck

A.4 GE U30C Truck

A.5 GE E60CP Truck

A.6 EMD GPSS Truck

This Appendix summarizes the truck test data presented in References 2 through 8. Some additional information, obtained from the truck manufacturers and References 10 through 13, is also presented to facilitate truck/locomotive dynamic modeling. If more detailed data are required, the References should be consulted*.

The data in this appendix are presented in terms of the "major" truck degrees of freedom usually included in an analytical model. Stiffness, friction, mass properties and geometric data are presented for the various trucks tested. Table A-1 describes the truck degrees of freedom for which data is presented.

Table A-1 Truck Degrees of Freedom for which Data is Presented

DEGREES OF FREEDOM	DESCRIPTION
Primary Vertical	Relative vertical motion between frame and wheelsets
Secondary Vertical	Relative vertical motion between bolster and frame
Primary Lateral	Relative lateral motion between frame and wheelsets
Secondary Lateral	Relative lateral motion between bolster and frame
Primary Yaw	Relative yaw motion between frame and wheelsets
Bolster Yaw	Relative yaw motion between carbody and bolster at bolster center plate

The following paragraphs present a summary of the measured characteristics of each truck tested. In addition, some representative locomotive carbody data is presented. Many of the trucks tested are available with a range of suspension stiffnesses to fit the particular service application. The measured values presented in this report, and the detailed test reports, only reflect the particular trucks tested. Any analyses, using the data obtained during testing, should include consideration of sensitivity of analysis results to variations in suspension parameters. Friction values are very dependent on truck cleanliness and wear of friction pads. Consequently, a range of friction values should be used in analytical simulations.

Geometric data is presented, based on the dimensional parameters shown in Figure A-1.

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*All mass property and geometric data are approximate.

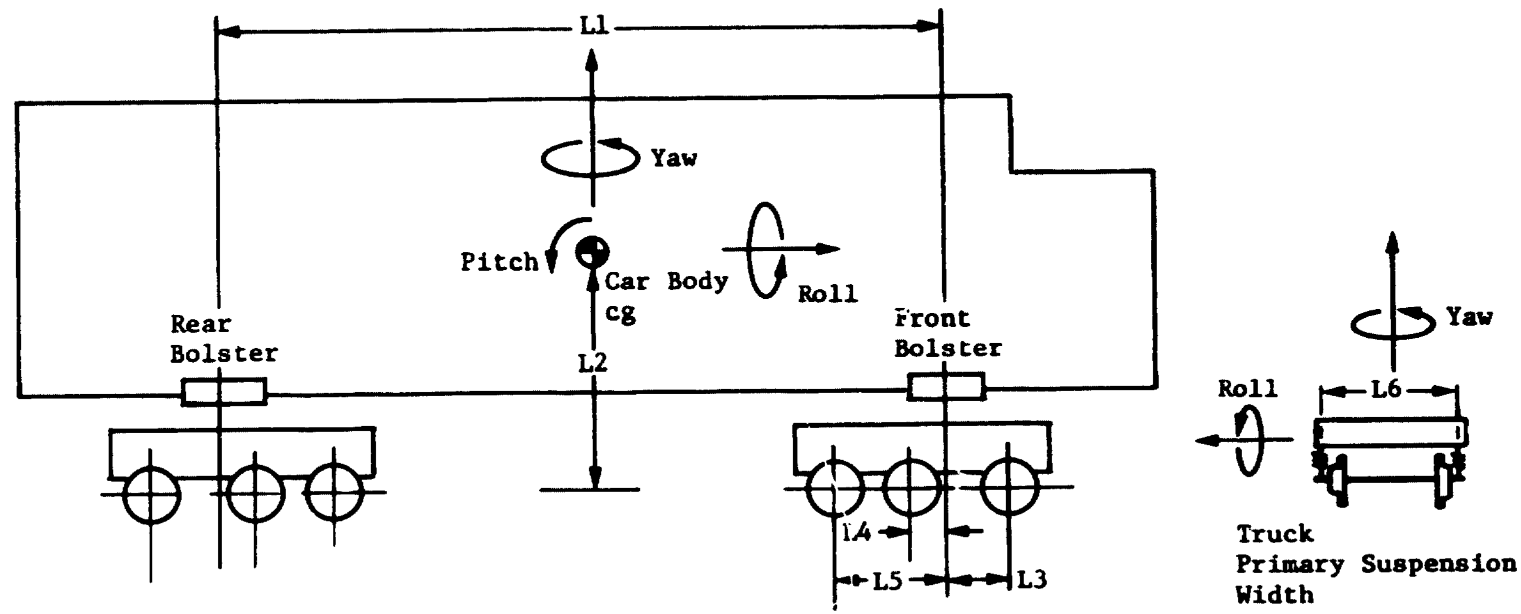

Figure A-1 Truck and Car Body Geometric Nomenclature

A.1 EMD E8 Truck

A.1 EMD E8 TRUCK

An early vintage (1940's) three-axle EMD truck (the center axle is not driven). The truck has a swing hanger secondary lateral suspension, with transverse leaf springs for the secondary vertical suspension. The primary vertical suspension has load equilization struts, which attempt to equalize vertical axle loads. No external dampers are employed on the truck. Figure 2-16 is a photograph of the E8 truck tested. Following is a summary of carbody and truck data for the E8 truck.

Table A-2 Locomotive Carbody Data for the E8 Truck

LOCOMOTIVE MODEL	MASS ($LB\text{-}S^2/IN.$) CARBODY + BOLSTERS	INERTIA ($LB\text{-}IN.\text{-}S^2$)			L1(IN.)	L2(IN.)
		ROLL	PITCH	YAW		
E8	622.5	1.07×10^6	2.52×10^7	2.52×10^7	516.	59.
E9*	622.5	1.07×10^6	2.52×10^7	2.52×10^7	516.	59.

*Figure A-2

Table A-3 E8 Truck Mass and Geometric Properties

COMPONENT	MASS ($LB\text{-}S^2/IN.$)	INERTIA ($LB\text{-}IN.\text{-}S^2$)		L3(IN.)	L4(IN.)	L5(IN.)	L6(IN.)
		ROLL	YAW				
Frame	40.	5.5×10^4	1.7×10^5				78.
Wheelset	30.3*,#		1.65×10^4	+84.**	0.	-84.	

*With motor (center axle is not driven, Mass ≈ 9.1 $LB\text{-}S^2/IN.$)
**Front and rear trucks are identical, + is forward.
#The unsprung wheelset mass is ≈ 22. $LB\text{-}S^2/IN.$

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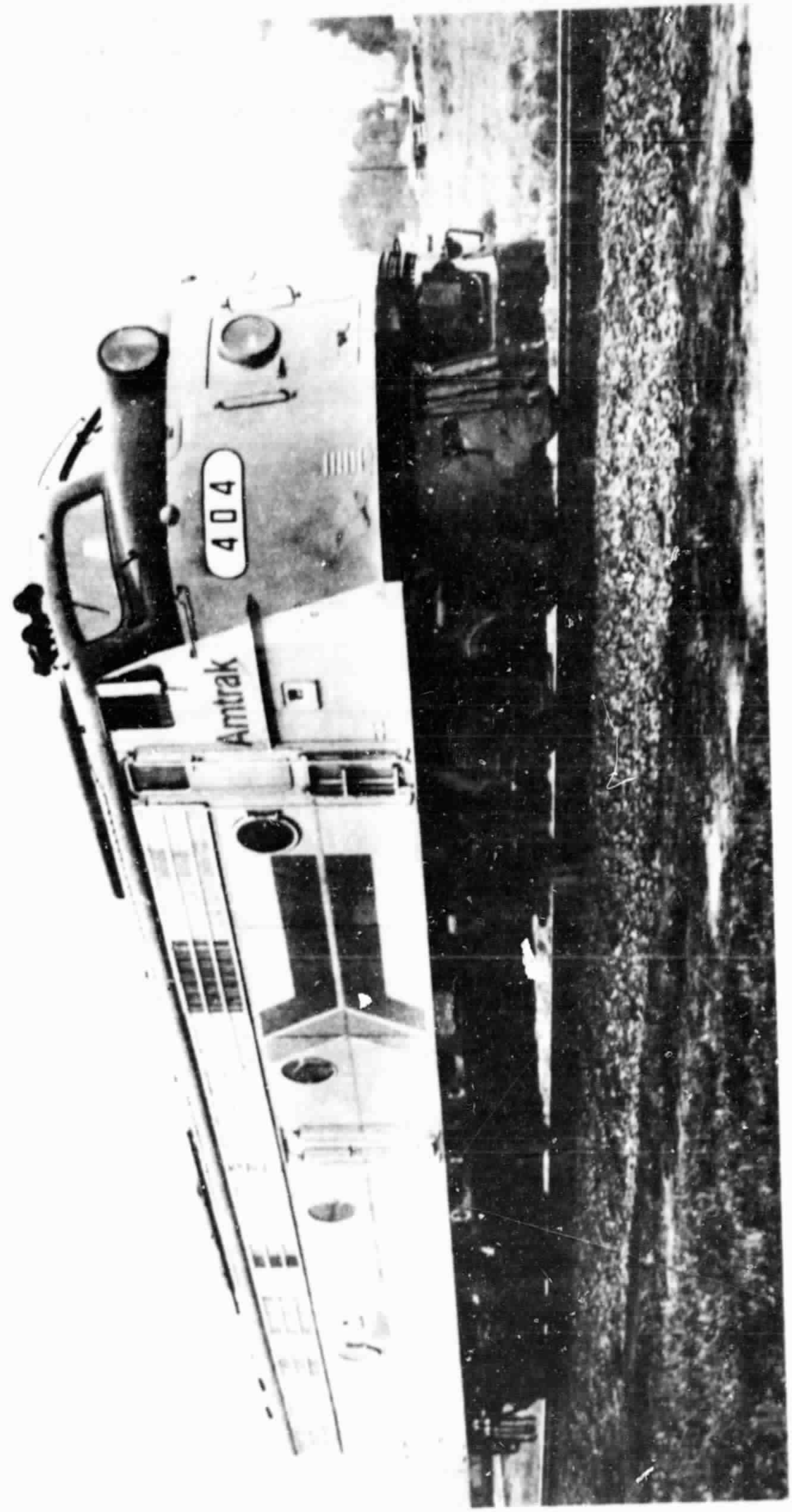
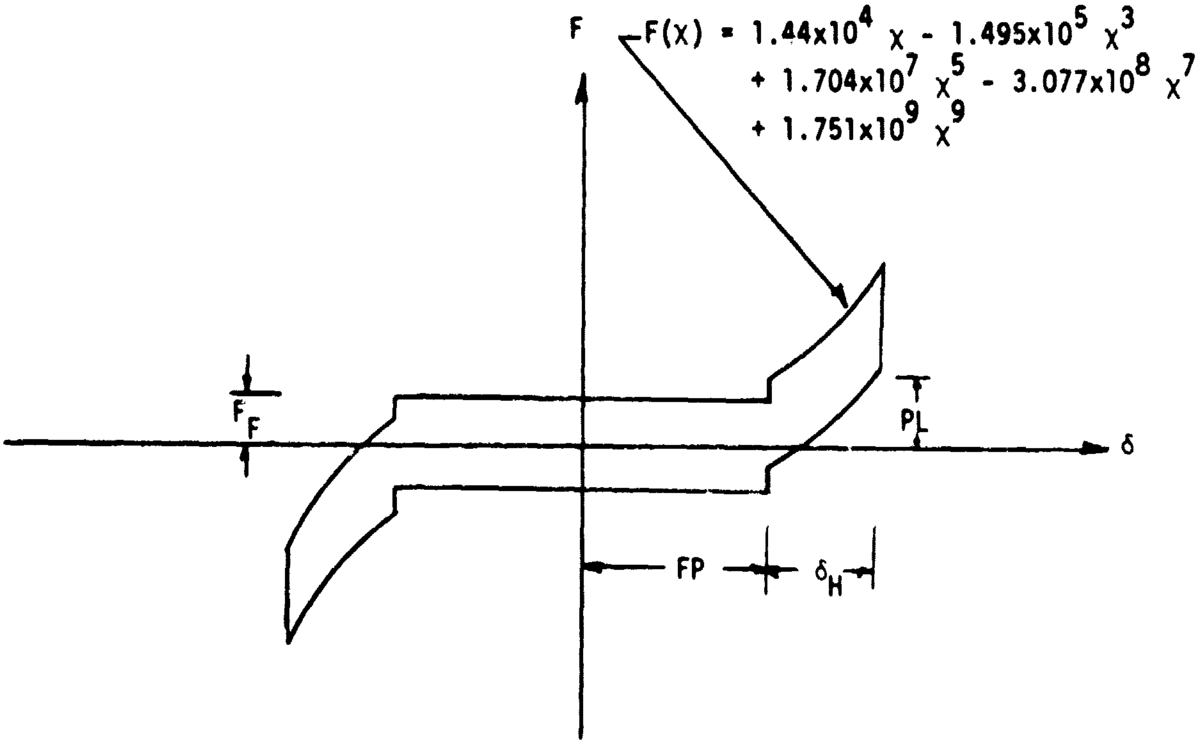


Figure A-2 EMD E9 Locomotive with E8 Trucks

Table A-4 E8 Truck Suspension Characteristics

SUSPENSION SYSTEM	STIFFNESS	FRICTION	COMMENT
Primary Vertical [Coil Springs]	70,000. LB/IN.	4,000. to 8,500. LB	
Secondary Vertical [Leaf Springs, 2 ea]	51,000. LB/IN.	13,000. to 17,000. LB	At 30,000. LB of tractive effort, the rear of the bolster did not move, due to friction.
Primary Lateral [Hyatt Bearing]	Figure A-3	Dynamic Coefficient of Friction ≃ 0.1	Lateral Freeplay, δ = 3/16"-3/8"
Secondary Lateral [Swing Hanger]	4,622. LB/IN. (@ w = 120,164. LB)	1,500. LB	Stiffness based on pendulum equation: $K = \frac{w}{\ell}$, $\ell \simeq 26"$
Primary Yaw [Longitudinal Axle Yaw]	Figure A-4	N/A	Polynomial stiffness due to compression of pedestal liners. Freeplay ≃ 1/8"-3/16".
Bolster Yaw	N/A	150,000. IN.-LB*	$T = \frac{2}{3} \mu F_N \frac{(R_o^3 - R_i^3)}{(R_o^2 - R_i^2)}$, IN.-LB where, μ = coefficient of friction (≃0.1) F_N = total normal force R_o = Outer radius of bolster plate (IN.) R_i = Inner radius of bolster plate (IN.)

*Calculated not measured.



where:
- FP = axle freeplay, 3/16"-3/8"
- δ_H = allowable compression of rubber bumper, ≈ 1/4"
- PL = bumper preload, 1,000.-2,000. LB
- F_F = lateral bearing friction force, LB

$$F_F = \mu \frac{W_{LOCO}}{N_A} \frac{V_{LAT}}{\left[\left(\frac{r_a V_{LOCO}}{R_W}\right)^2 + V_{LAT}^{2}\right]^{1/2}}$$

- μ = coefficient of friction, ≈ 0.1
- W_{LOCO} = locomotive weight, LB
- N_A = total number of locomotive axles (4 or 6)
- V_{LAT} = relative axle/frame lateral velocity, IN./S
- r_a = axle radius at bearing, ≈ 3.2"
- R_W = wheel radius, IN.
- V_{LOCO} = locomotive forward velocity, IN./S

$X = \delta - FP + \delta_o$

δ_o = rubber bumper installed preload compression, IN.

Figure A-3 Primary Lateral Load Deflection Characteristics

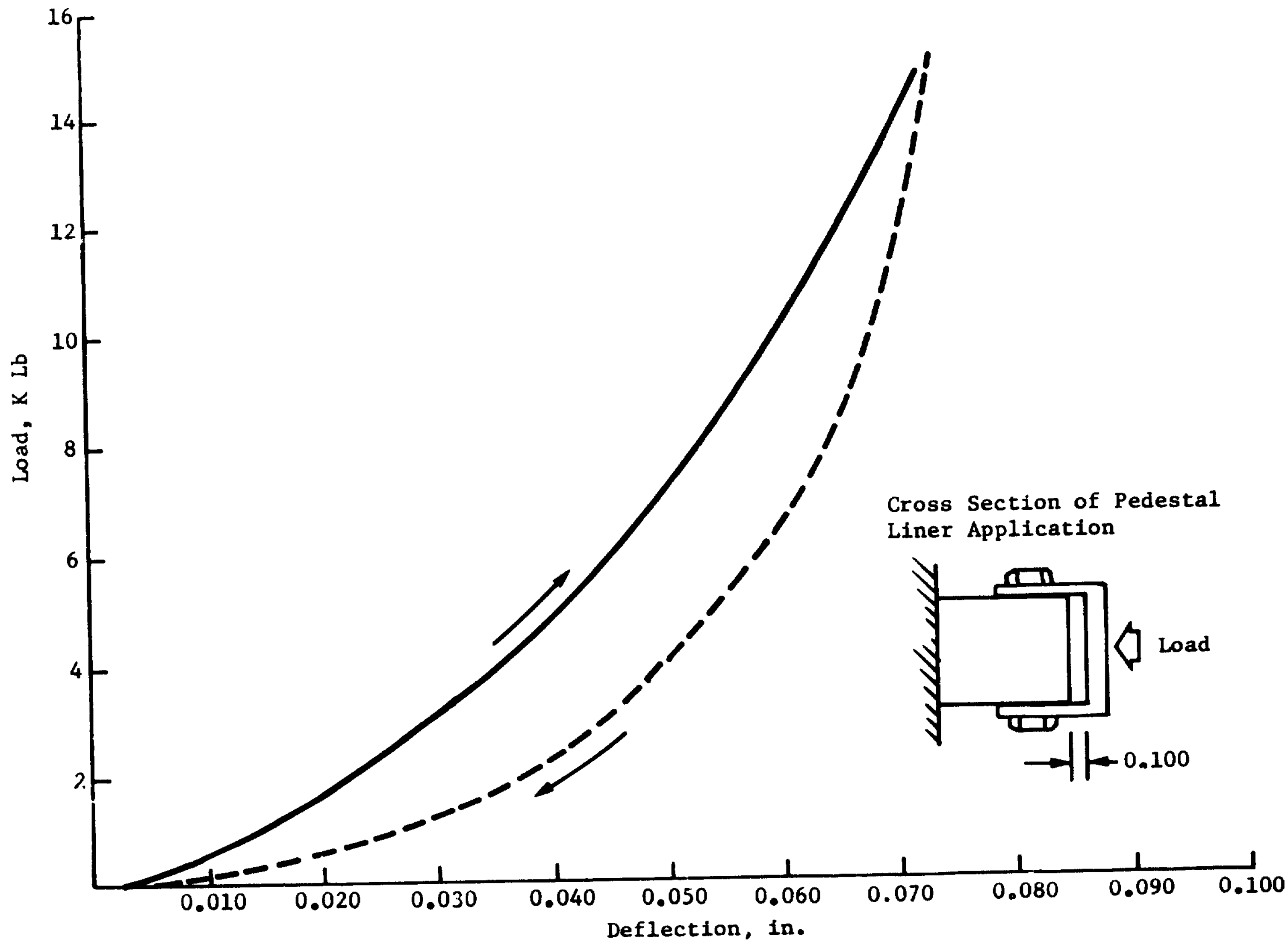


Figure A-4 Longitudinal Load-Deflection Curve for Typical Nylatron Pedestal Liner Application (Test by EMD 10-7-76, K.R. Smith)

A.2 EMD Flexicoil Truck

A.2 EMD Flexicoil Truck

An EMD three-axle design produced for SD type locomotives, prior to development of the HT-C truck design. The truck employs a standard secondary suspension with coil springs as the active suspension element. Two hydraulic shock absorbers are used in the primary vertical suspension system. They are located on the center axle. Figure 2-17 is a photograph of the Flexicoil truck during testing. Following is a summary of carbody and truck data for the Flexicoil truck.

Table A-5 Locomotive Carbody Data for the Flexicoil Truck

LOCOMOTIVE MODEL	MASS ($LB-S^2/IN.$) CARBODY + BOLSTERS	INERTIA ($LB-IN.-S^2$)			L1(IN.)	L2(IN.)
		ROLL	PITCH	YAW		
SD38	668.	1.6×10^6	3.1×10^7	3.4×10^7	480.	59.
SD40	678.7	1.6×10^6	3.2×10^7	3.4×10^7	480.	59.
SD45	678.7	1.6×10^6	3.2×10^7	3.4×10^7	480.	59.

Table A-6 Flexicoil Truck Mass and Geometric Properties

COMPONENT	MASS ($LB-S^2/IN.$)	INERTIA ($LB-IN.-S^2$)		L3(IN.)	L4(IN.)	L5(IN.)	L6(IN.)
		ROLL	YAW				
Frame	36.	5.2×10^4	1.63×10^5				79.
Wheelset	30.3**		1.65×10^4	+81.5*	0.	-81.5	

*Front and rear trucks are identical, + is forward.
**The unsprung wheelset mass is ≈ 22. $LB-S^2/IN.$

Table A-7 Flexicoil Truck Suspension Characteristics

SUSPENSION SYSTEM	STIFFNESS	FRICTION	COMMENTS
Primary Vertical [Coil Springs]	53,870. LB/IN.	0.16 TE*, LB	
Secondary Vertical [Coil Springs, 4 ea]	50,000. LB/IN.	0.5 TE, LB	Secondary susepnsion contains a preloaded friction device; initial friction ≃ 2,800. LB.
Primary Lateral [Hyatt Bearing]	Figure A-3	Dynamic coefficient of friction ≃ 0.1	Lateral Freeplay, δ = 3/16"-3/8"
Secondary Lateral [Floating Bolster on Coil Springs]	18,000. LB/IN.	0.4 TE, LB	See secondary vertical above. Initial friction ≃ 2,800. LB.
Primary Yaw [Longitudinal Axle Yaw]	Figure A-4	N/A	Polynomial stiffness due to compression of pedestal liners. Free Play ≃ 1/8" - 3/16".
Bolster Yaw	N/A	125,000. IN.-LB	Center plate lubricated with oil.

*TE = Tractive Effort, LB.

A.3 EMD HT-C Truck

A.3 EMD HT-C Truck

This three-axle EMD design appeared on the market in the early 1970's. HT-C stands for High Traction-Three Axle. This truck also employs a standard secondary suspension utilizing single segment rubber pad springs as the active suspension elements. Two hydraulic shock absorbers are used on the center axle of the primary vertical suspension system. Early operational derailment problems led to design modifications including: the use of softer rubber pad springs and the addition of a hydraulic shock absorber in the secondary lateral suspension. Both versions of the truck were tested. Figure 2-18 is a photograph of the first HT-C truck tested. The second truck was tested at both ambient and cold temperatures. Following is a summary of carbody and truck data for the HT-C trucks.

Table A-8 Locomotive Carbody Data for the HT-C Truck

LOCOMOTIVE MODEL	MASS ($LB\text{-}S^2/IN.$) CARBODY + BOLSTERS	INERTIA ($LB\text{-}IN.\text{-}S^2$)			L1(IN.)	L2(IN.)
		ROLL	PITCH	YAW		
FREIGHT						
SD38-2	669.-804.	1.51×10^6	35.3×10^6	35.3×10^6	522.	57.7
SD40-2	669.-804.	1.51×10^6	35.3×10^6	35.3×10^6	522.	57.7
SD45-2	669.-804.	1.51×10^6	35.3×10^6	35.3×10^6	522.	57.7
PASSENGER						
SDP40F*	742.-766.	1.72×10^6	39.6×10^6	39.6×10^6	552.	57.7

*Figure A-5.

Table A-9 HT-C Truck Mass and Geometric Properties

COMPONENT	MASS ($LB\text{-}S^2/IN.$)	INERTIA ($LB\text{-}IN.\text{-}S^2$)		L3(IN.)	L4(IN.)	L5(IN.)	L6(IN.)
		ROLL	YAW				
Frame	40.	5.27×10^4	1.61×10^5				79.
Wheelset	34.#		1.64×10^4	+78.4*	-1.25*	-85.0*	

*Rear truck is same as front truck rotated 180^0 about bolster center; signs are reversed (+ is forward).
#The unsprung wheelset mass is ≃ 22. $LB\text{-}S^2/IN$.

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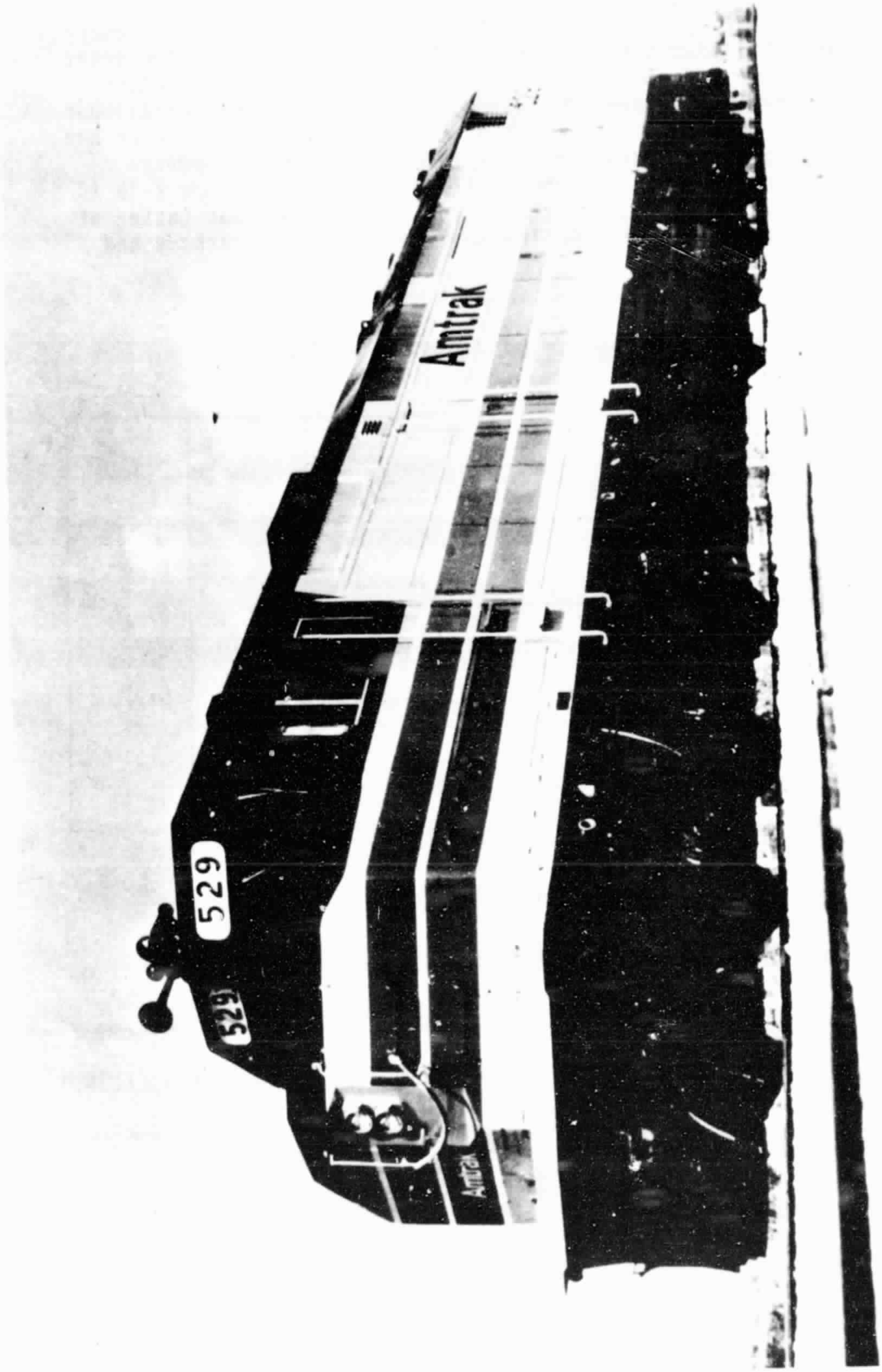

Figure A-5 EMD SDP40F Locomotive with HT-C Trucks

Table A-10 HT-C Truck Suspension Characteristics#

SUSPENSION SYSTEM	STIFFNESS		FRICTION		COMMENTS
	ORIGINAL DESIGN	MODIFIED DESIGN	ORIGINAL DESIGN	MODIFIED DESIGN	
Primary Vertical [Coil Springs]	44,000. LB/IN.	33,800. LB/IN.	0.16 TE*, LB	0.35 TE, LB(a)** 0.43 TE, LB(c)	
Secondary Vertical [Single Segment Rubber Pad Springs, 4ea]	475,000. LB/IN.	408,000. LB/IN.(a) 571,000. LB/IN.(c)	0.5 TE, LB	0.42 TE, LB(a) 0.7 TE, LB(c)	
Primary Lateral [Hyatt Bearings]	Figure A-3		Dynamic coefficient of friction ≈ 0.1		Lateral Freeplay, δ = 3/16"-3/8"
Secondary Lateral [Floating bolster on single segment rubber pad springs]	23,000. LB/IN.	15,500. LB/IN.(a) 17,500. LB/IN.(c)	0.4 TE, LB	0.4 TE, LB(a) 0.44 TE, LB(c)	
Primary Yaw [Longitudinal Axle Yaw]	Figure A-4		N/A		Polynomial stiffness due to compression of pedestal liners. Freeplay ≈ 1/8"-3/16"
Bolster Yaw	N/A		150,000. IN.-LB(a) 105,000. IN.-LB(c)		Center plate lubricated with oil. Test performed on modified design only.

#Modified design tested at ambient and cold temperatures.
*TE = Tractive Effort, LB.
**(a) = Ambient (≈70°F)
(c) = Cold (≈0°F)

A.4 GE U30C Truck

A.4 GE U30C Truck

A three-axle GE truck produced for diesel electric service. This truck has a standard secondary suspension with five-segment rubber pad springs as the active suspension elements. Friction snubbers are used to provide primary vertical suspension damping. Early versions of this truck employed four snubbers (front and rear axles). Later versions use only two snubbers (on the center axle). Figure 2-19 shows the U30C truck during testing. Following is a summary of carbody and truck data for the U30C truck.

Table A-11 Locomotive Carbody Data for the U30C Truck

LOCOMOTIVE MODEL	MASS ($LB\text{-}S^2/IN.$) CARBODY + BOLSTERS	INERTIA ($LB\text{-}IN.\text{-}S^2$)			L1(IN.)	L2(IN.)
		ROLL	PITCH	YAW		
U33C	685.-831.	1.72×10^6	39.6×10^6	39.6×10^6	491.	50.3
C30-7*	691.-831.	1.72×10^6	39.6×10^6	39.6×10^6	491.	50.3
C36-7	691.-831.	1.72×10^6	39.6×10^6	39.6×10^6	491.	50.3

*Figure A-6.

Table A-12 U30C Truck Mass and Geometric Properties

COMPONENT	MASS ($LB\text{-}S^2/IN.$)	INERTIA ($LB\text{-}IN.\text{-}S^2$)		L3(IN.)	L4(IN.)	L5(IN.)	L6(IN.)
		ROLL	YAW				
Frame	37.6	5.6×10^4	1.78×10^5				79.
Wheelset	30.3#		1.65×10^4	+79.5*	+2.0*	-83.5*	

*Rear truck is same as front, + is forward.
#The unsprung wheelset mass is ≈ 22. $LB\text{-}S^2/IN.$

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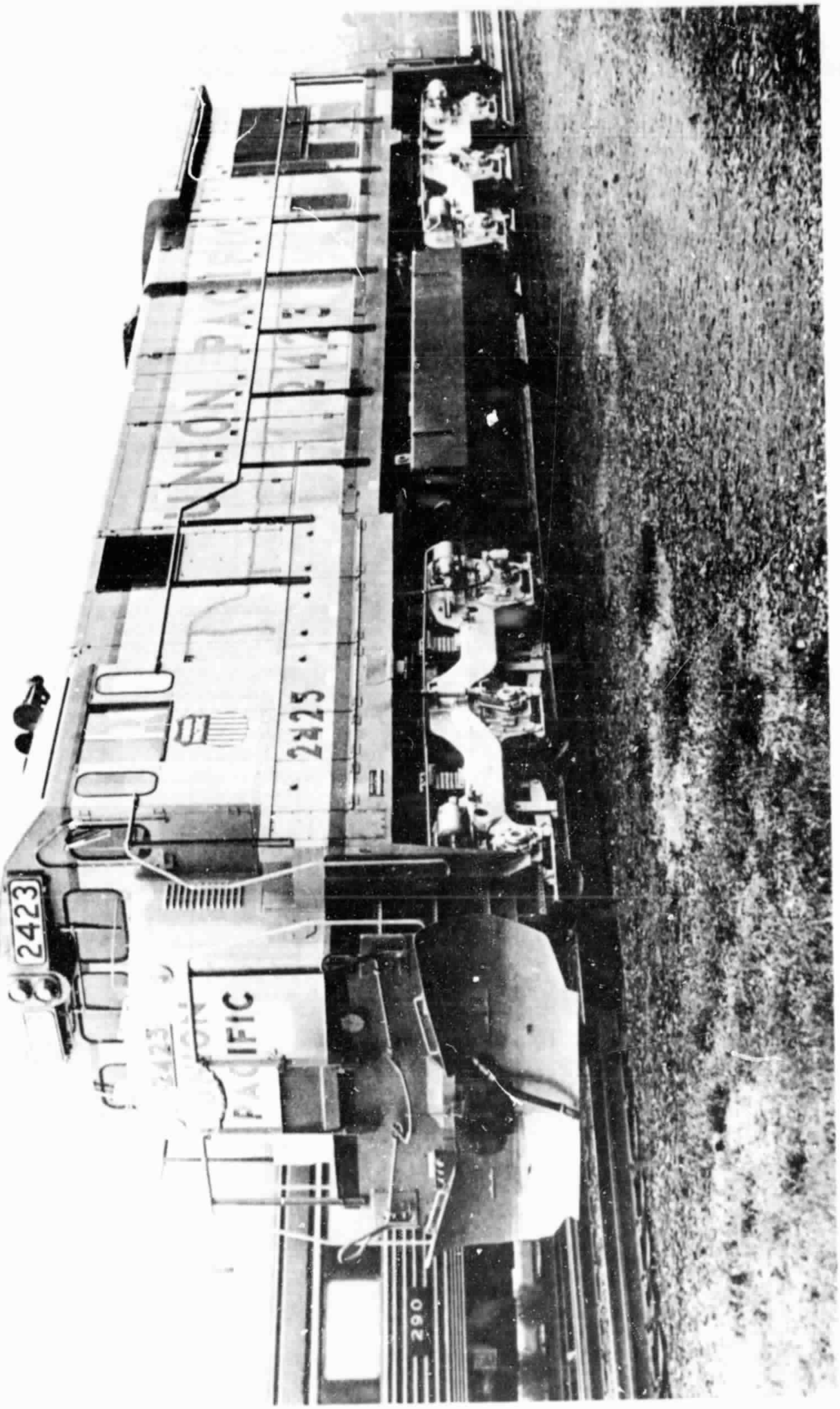

Figure A-6 GE C30-7 Locomotive with U30C Trucks

Table A-13 U30C Truck Suspension Characteristics

SUSPENSION SYSTEM	STIFFNESS	FRICTION	COMMENTS
Primary Vertical [Coil Springs]	47,500. LB/IN.	0.16 TE*, LB	
Secondary Vertical [Five Segment Rubber Pad Springs, 4 ea]	467,000. LB/IN.	0.45 TE, LB	
Primary Lateral [Hyatt or Timken Bearings]	Figure A-3 (Hyatt Bearing)	Dynamic Coefficient of Friction ≈ 0.1	Lateral Freeplay, δ = 3/16"-3/8"
Secondary Lateral [Floating bolster on five segment rubber pad springs]	12,000. LB/IN.	0.5 TE, LB	
Primary Yaw [Longitudinal Axle Yaw]	Figure A-4	N/A	Polynomial stiffness due to compression of pedestal liners. Freeplay ≈ 1/8" - 3/16".
Bolster Yaw	N/A	140,000. IN.-LB	Center plate lubricated with oil.

*TE = Tractive Effort, LB.

A.5 GE E60CP Truck

A.5 GE E60CP Truck

A three-axle GE truck produced for all-electric Amtrak service. The basic design of this truck is essentially the same as the U30C truck, with the exception of external damping devices. This truck employs eight hydraulic shock absorbers: four in the primary vertical suspension (front and rear axles), two between the truck frame and locomotive carbody to damp yaw motion, and two in the secondary lateral suspension. The truck configuration is similar to the U30C design shown in Figure 2-19. Following is a summary of carbody and truck data for the E60CP truck.

Table A-14 Locomotive Carbody Data for the E60CP Truck

LOCOMOTIVE MODEL	MASS ($LB-S^2/IN.$) CARBODY + BOLSTERS	INERTIA ($LB-IN.-S^2$)			L1(IN.)	L2(IN.)
		ROLL	PITCH	YAW		
E60CP*	670.	1.31×10^6	$32. \times 10^6$	$32. \times 10^6$	540.	52.3

*Figure A-7.

Table A-15 E60CP Truck Mass and Geometric Properties

COMPONENT	MASS ($LB-S^2/IN.$)	INERTIA ($LB-IN.-S^2$)		L3(IN.)	L4(IN.)	L5(IN.)	L6(IN.)
		ROLL	YAW				
Frame	40.7	5.26×10^4	1.78×10^5				79.
Wheelset	34.1#		2.13×10^4	+79.5*	+2.0	-83.5	

*Rear truck is same as front, + is forward.
#The unsprung wheelset mass is ≈ 22. $LB-S^2/IN.$

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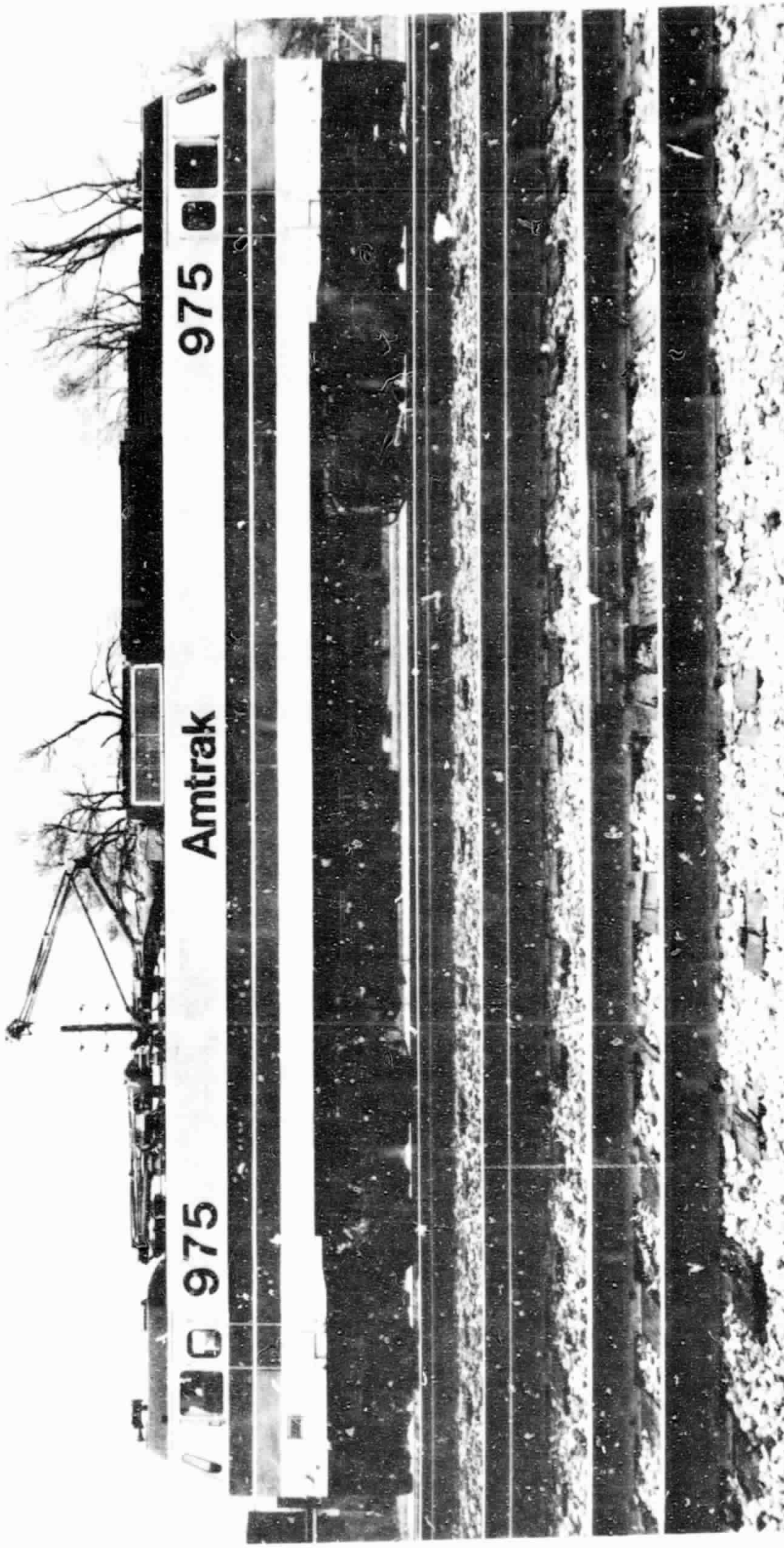

Figure A-7 GE E60CP Locomotive with E60CP Trucks

Table A-18 E60CP Truck Suspension Characteristics

SUSPENSION SYSTEM	STIFFNESS	FRICTION	COMMENTS
Primary Vertical [Coil Springs]	34,400. LB/IN.	0.17 TE*, LB	
Secondary Vertical [Five segment rubber pad springs, 4 ea]	542,000. LB/IN.	0.41 TE, LB	
Primary Lateral [Hyatt or Timken Bearings]	Figure A-3 (Hyatt Bearing)	Dynamic Coefficient of Friction ≃ 0.1	Lateral Freeplay, δ = 3/16" - 3/8"
Secondary Lateral [Floating bolster on five segment rubber pad springs]	10,600. LB/IN.	0.2 TE, LB	
Primary Yaw [Longitudinal Axle Yaw]	Figure A-4	N/A	Polynomial stiffness due to compression of pedestal liners. Freeplay ≃ 1/8" - 3/16"
Bolster Yaw	N/A	190,000. IN.-LB	Center plate lubricated with oil.

*TE = Tractive Effort, LB.

A.6 EMD GPSS Truck

A.6 EMD GPSS Truck

A two-axle EMD truck. The truck has a swing hanger secondary lateral suspension, with rubber pad springs for the secondary vertical suspension. Two versions of this truck were tested: the original version having rubber pad springs in compression, and the modified design having inclined rubber pad springs. The inclined rubber pad springs result in a slightly softer vertical suspension. Two hydraulic shock absorbers are used in the primary vertical suspension system (diagonally opposite on the front and rear axles). Figure 2-20 is a photograph of the original version GPSS truck tested. The two segment rubber pad springs are visible in the photograph. Figure 2-21 is a closeup of the inclined rubber pad springs in the modified truck (the second GPSS truck tested). Both trucks were tested at ambient and cold temperatures. Following is a summary of carbody and truck data for the GPSS trucks.

Table A-17 Locomotive Carbody Data for the GPSS Truck

LOCOMOTIVE MODEL	MASS (LB-S^2/IN.) CARBODY + BOLSTERS	INERTIA (LB-IN.-S^2)			L1(IN.)	L2(IN.)
		ROLL	PITCH	YAW		
FREIGHT						
GP38-2	462.	1.6×10^6	1.2×10^7	1.2×10^7	408.	60.
GP39-2	462.	1.6×10^6	1.2×10^7	1.2×10^7	408.	60.
GP40-2	477.5	1.6×10^6	1.3×10^7	1.3×10^7	408.	60.
GP50	487.9	1.6×10^6	1.3×10^7	1.3×10^7	408.	60.
PASSENGER						
F40PH*,#	476.3-494.4	1.6×10^6	1.3×10^7	1.3×10^7	396.	60.

*Inclined secondary suspension rubber pad prings used on F40PH only.
#Figure A-8.

Table A-18 GPSS Truck Mass and Geometric Properties

COMPONENT	MASS (LB-S^2/IN.)	INERTIA (LB-IN.-S^2)		L3(IN.)	L4(IN.)	L5(IN.)	L6(IN.)
		ROLL	YAW				
Frame	29.8	3.86×10^4	13.15×10^4				79.
Wheelset	30.#		2.13×10^4	+54.*	0.	-54.*	
Swing Hanger	2.97	2.1×10^3	2.2×10^3				

*Rear truck same as front, + is forward.
#The unsprung wheelset mass is ≃ 22. LB-S^2/IN.

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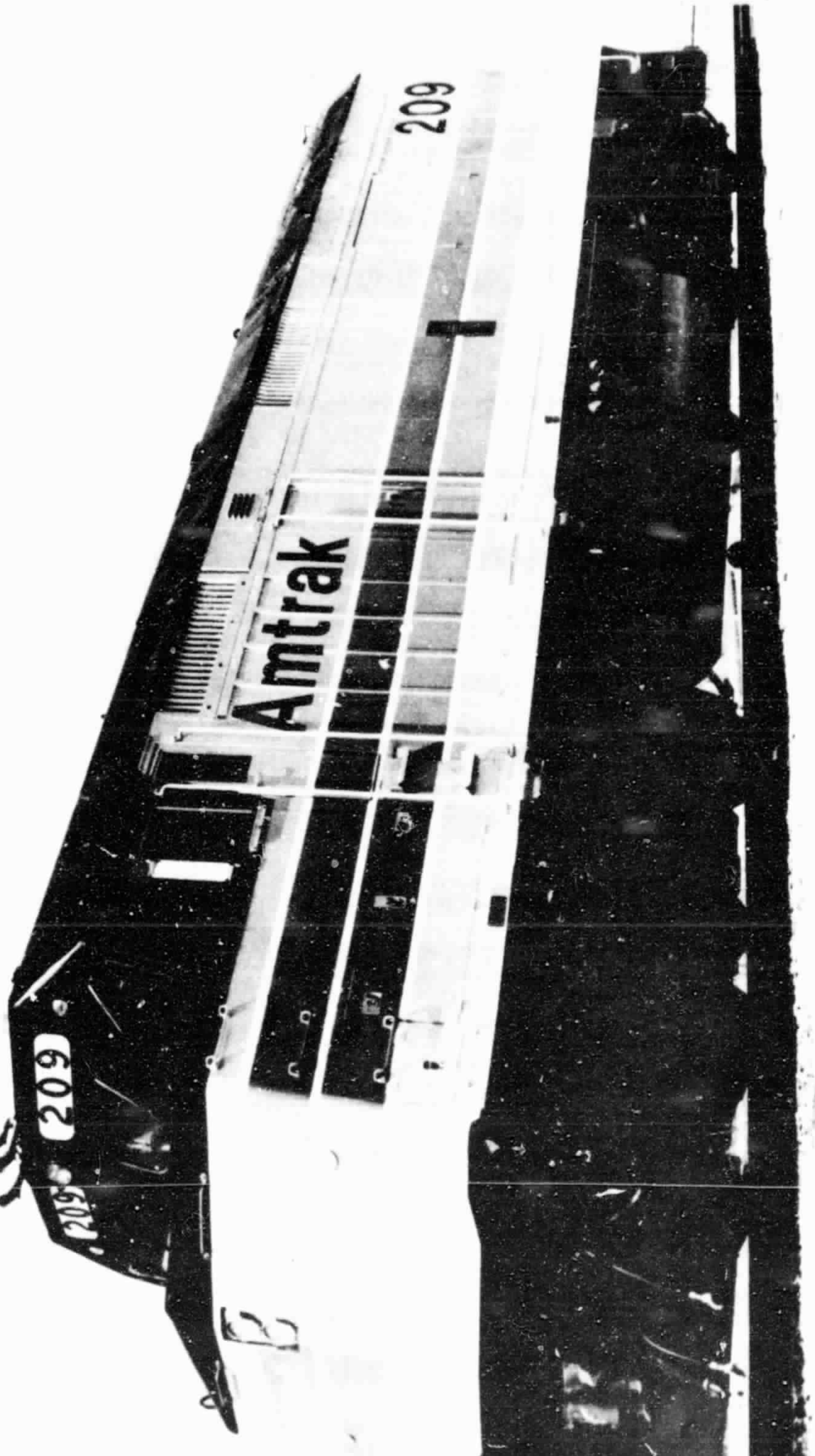

Figure A-8 EMD F40PH Locomotive with GPSS Trucks (2 Axle)

Table A-19 GPSS Truck Suspension Characteristics

SUSPENSION SYSTEM	STIFFNESS		FRICTION		COMMENTS
	ORIGINAL DESIGN	MODIFIED DESIGN	ORIGINAL DESIGN	MODIFIED DESIGN	
Primary Vertical [Coil Springs]	34,100. LB/IN. (a,c)*		0.16 TE-0.8 TE,** LB (a,c)		
Secondary Vertical [Rubber Pad Springs]	187,700. LB/IN. (a) 245,600. LB/IN. (c)	86,900. LB/IN. (a) 104,700. LB/IN. (c)	0.5 TE -2.0 TE, LB (a,c)		The GPSS truck has two two-segment rubber pad springs. The modified design has four four-segment inclined rubber pad springs.
Primary Lateral [Hyatt Bearings]	Figure A-3		Dynamic Coefficient of Friction ≃ 0.1		Lateral Freeplay, δ = 3/16"-3/8"
Secondary Lateral [Swing Hanger]	3,856. LB/IN. ($\ell \simeq$ 24.75 IN.)	3,567.7 LB/IN. ($\ell \simeq$ 26.75 IN.) (@ w = 95,436.5 LB)	0.5 TE-2.0 TE, LB (a,c)		Stiffness based on pendulum equation: $K = \frac{w}{\ell}$
Primary Yaw [Longitudinal Axle Yaw]	Figure A-4		N/A		Polynomial stiffness due to compression of pedestal liners. Freeplay ≃ 1/8"-3/16"
Bolster Yaw	N/A		Dry 148-179 (a) 168-237 (c)	Lubricated 118-139 (a) 181-223 (c)	IN-LBx10^{-3}

* a = Ambient (≃70°F), c = Cold (≃0°F).
** TE = Tractive Effort, LB.

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